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BRUCE NGS'B' - BOILER FEED PUMP MOTOR VIBRATION AND ROTOR DYNAMICS ANALYSES

Report No 87-246-K

F. Perricone,
Unit Head, Structural Dynamics
N.A. Jagan, Engineer
Plant Equipment Dynamics Section
Mechanical Research Department

ABSTRACT

An experimental vibration analysis and a theoretical rotor dynamics analysis were carried out to determine the causes of high vibration levels. The results show that the motor has a rotor critical speed near the operating speed resulting in amplification of the rotor vibration. The theoretical rotor dynamics analysis further indicated that the rotor response is sensitive to the bearing clearances. It is recommended that an investigation be carried out to determine practical bearing and/or structural modifications that can be implemented to sufficiently move the rotor critical speed away from the operating speed.

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EXECUTIVE SUMMARY

BRUCE NGS'B' - BOILER FEED PUMP MOTOR VIBRATION AND ROTOR DYNAMICS ANALYSES

F. Perricone and N.A. Jagan
Plant Equipment Dynamics Section
Mechanical Research Department

The Boiler Feed Pump Motors have a history of high vibration levels which have resulted in an above average amount of maintenance. Experimental and theoretical vibration analyses were carried out to determine the causes of the high vibration levels.

The experimental investigation on a typical motor (Unit #5, Boiler Feed Pump Motor #1) revealed that the motor most probably operates with a rotor critical speed near the operating speed. A rotor dynamics analysis confirmed the experimental results. Amplification of rotor vibration due to operation near the first rotor critical speed is the main cause of the high motor vibration levels.

It is recommended that an investigation be carried out to determine practical bearing and/or structural modifications that can be implemented to detune the rotor-bearing-support system and thereby significantly reduce rotor amplification.

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To Mr. J. Bagshaw
Technical Manager
Bruce NGS'B' Operations

BRUCE NGS'B' - BOILER FEED PUMP MOTOR VIBRATION AND ROTOR DYNAMICS ANALYSES

1.0 INTRODUCTION

The Boiler Feed Pump Motors have a history of high vibration levels which has resulted in several bearing failures and an above average amount of maintenance. Research was requested to:

- Carry out a vibration survey and experimentally investigate the rotor and structural dynamics of a typical motor - Unit #5, Boiler Feed Pump Motor #1.

- Perform a theoretical rotor dynamics analysis to assess the dynamic characteristics of the existing rotor-bearing-support system.

The motor is a 4178 kW (5600 HP), 4160V, 3569 rpm, two pole induction motor equipped with journal bearings.

2.0 MEASUREMENT LOCATIONS, INSTRUMENTATION AND TEST PROGRAM

2.1 Measurement Locations

Two displacement transducers (90° apart) as well as vertical and horizontal accelerometers were installed on the motor inboard and outboard bearings. Also, one accelerometer was installed on the outboard bearing in the axial direction. A phase transducer was installed on the inboard bearing to provide the motor speed.

2.2 Measurements

2.2.1 Overall Vibration Levels

Bearing casing overall velocity levels were measured using a Bruel & Kjaer Model 4368 accelerometer, a Bruel & Kjaer Model 2635 charge amplifier and a Bently Nevada Digital Vector Filter (DVF2). Rotor displacement overall levels were obtained using the Bently Nevada 7200 series non-contact displacement transducer system and the DVF2.

2.2.2 Velocity Spectra

Velocity spectra were generated by analyzing the acceleration signal obtained from the Bruel & Kjaer Model 2635 charge amplifier with an Ono Sokki CF920 dual channel FFT analyzer.

2.2.3 Displacement Spectra

Displacement spectra were obtained by analyzing the displacement signal from the Bently Nevada 7200 series displacement transducer systems with an Ono Sokki CF920 dual channel FFT analyzer.

2.2.4 Sweep Spectra

Rotor and structural natural frequencies were determined by sine sweep testing. The motor was excited throughout a range of frequencies (5 - 150 Hz) with a 454 kg (1000 lb) hydraulic shaker instrumented with a force transducer. The shaker was mounted on the motor outboard bearing structure. For the structural dynamics analyses, the input force (shaker) and response (accelerometer) signals were fed simultaneously to a dual channel FFT analyzer which was used to calculate the input and response spectra as well as inertance and compliance transfer functions. For the rotor dynamics analyses, the input force (shaker) and response (relative and absolute displacement) signals were fed simultaneously to the dual channel FFT analyzer which was used to calculate the compliance transfer function. The validity of the transfer function was checked by determining the coherence of the input and response signals. A coherence value greater than 0.8 indicates that the response is predominantly due to the input.

2.3 Instrumentation

The main components of the instrumentation used in obtaining the measurements are listed in Table 1. All the signals were tape recorded for future analysis.

2.4 Test Program

The following tests were carried out.

2.4.1 Sweep Tests

Shaker sweep tests were carried out with the motor operating and not operating.

Although the primary purpose of the sweep tests was to identify the rotor natural frequencies, they were also used to determine the structural natural frequencies.

2.4.2 Running Test

Overall velocity and displacement levels as well as velocity and displacement spectra were obtained to determine the magnitude of vibration and spectral content under coupled and uncoupled conditions.

2.4.3 Peakhold Rundown Test

While in the uncoupled mode, peakhold rundown tests were carried out to assist in determining the location of rotor and structural natural frequencies.

3.0 EXPERIMENTAL RESULTS

3.1 Overall Vibration Levels and Vibration Spectra

The overall displacement and velocity levels are summarized in Tables 2 and 3.

The maximum relative displacement level of the motor rotor was measured at the outboard bearing in the horizontal direction (75-77 µm p-p). Rotor displacement for this type of machine and speed class generally should not exceed 50 µm p-p. The main displacement component is at 1 x rpm (Figures 1-4).

The maximum casing velocity levels were measured on the inboard bearing vertical direction (7.2 - 8.0 mm/s and the outboard bearing axial direction (7.7 - 8.5 mm/s). These vibration levels plotted on a general machinery severity chart (Figure 10) indicate that the levels are near the ROUGH severity level. These levels also exceed the limit of acceptance of 5 mm/s adopted by Ontario Hydro's Bruce NGS'B' nuclear power plant (Appendix B)/1/. Corrective actions to reduce these levels are recommended.

3.2 Dynamic Characteristics

3.2.1 Motor Rotor Natural Frequencies

Figure 11 shows a compliance transfer function (absolute rotor displacement/force) and phase obtained on the inboard bearing in the horizontal direction during a shaker sweep test with the motor operating. Operation of the motor during this test would ensure a fully-developed oil film in the journal during the

test. The results show a peak in the compliance transfer function at 53.42 Hz which is related to a rotor natural frequency near 1 X rpm (59.5 Hz).

The high vibration level at 1 X rpm measured on the bearing housings, as well as on the rotor is most likely due to a rotor natural frequency amplifying vibration at 1 X rpm.

3.3 Peakhold Rundown Spectrum

A peakhold spectrum of rotor displacement during rundown obtained, just after the power was shut off, on the motor outboard bearing horizontal direction is shown in Figure 12. This figure shows a peak in rotor displacement at about 53.5 Hz. This is most likely due to the presence of a rotor natural frequency close to 1 X rpm.

The absence of a sharper peak and higher displacement levels at the rotor first critical speed at about 53.5 Hz is most likely due to the fast rundown of the motor through the rotor natural frequency which does not allow the vibration amplitudes to build up.

4.0 ROTOR DYNAMICS ANALYSIS-EXISTING JOURNAL BEARING

4.1 Rotor-Bearing-Support Analytical Model

The rotor model used for the theoretical analysis is shown in Figure 13. The rotor dimensions and weights were provided by Bruce NGS'B'. The rotor is subdivided into 34 mass stations with additional weights near the centre of the rotor and at the fan locations representing the core and fan masses respectively.

The bearing fluid films are represented by both orthogonal and cross-coupled stiffnesses and damping coefficients.

The following journal bearing parameters were used in the analysis:

- bearing length: 83.82 mm (3·3 in)
- journal diameter: 139.7 mm + 0.0 µm (5.5 in + 0.0 in
  - 12.7 µm (- 0.0005 in)
- bearing diametral clearances: 152.4 µm (0.006 in) to
  330.2 µm (0.013 in)
  inclusive
- bearing load: 1247.4 kg (2750 lbs) on each bearing

- average lubricant temperature: 38°C

- lubricant viscosity @38°C: 2.756×10^{-2} Pa·s ($4.0 \times 10^{-6} \frac{lb \cdot s}{in^2}$)

Although the typical journal bearing diametral clearance measured in the field was approximately 203.2 μm (0.008 in), a range of bearing clearances from 152.4 μm (0.006 in) to 330.2 μm (0.013 in) in increments of 25.4 μm (0.001 in) was considered.

The bearing casing support stiffnesses used in the analysis were determined experimentally. The bearing support structure was excited through a range of frequencies (20 - 150 Hz) with a hydraulic shaker instrumented with a force transducer. The input force (shaker) and response (accelerometer signal integrated twice to yield displacement) were fed simultaneously into a dual channel FFT analyzer which calculated the compliance transfer function. The horizontal and vertical bearing support stiffnesses were determined from these compliance transfer functions. The bearing casing support stiffnesses used in the rotor dynamics analysis are:

- horizontal stiffness: 4.90×10^8 N/m (2.80×10^6 lb/in)

- vertical stiffness: 3.63×10^8 N/m (2.07×10^6 lb/in)

The damping in the bearing casing support was assumed to be negligible. The dynamic mass of the support, used in the analysis, was 227 kg (500 lbs).

4.2 Bearing Oil Film Stiffness and Damping Coefficients

The journal bearing oil film stiffness and damping coefficients were determined with "FINBRG1", a fluid film bearing computer program developed by the University of Virginia/2/. The orthogonal co-ordinate sign convention used in "FINBRG1" is shown in Figure 14. The analytical results for the bearing stiffness and damping coefficients for the eight bearing diametral clearances are summarized in Table 4.

4.3 Rotor Unbalance Response Analysis

The synchronous unbalance response analysis determines the damped response of the rotor to an assumed unbalance distribution on the rotor. From this analysis the damped rotor critical speeds, the sensitivity of the machine to unbalance and characteristic rotor vibration levels are determined. Since the

bearing stiffnesses are of the same order of magnitude as the bearing support stiffnesses, the bearing support flexibility was considered in the analysis. The rotor unbalance response analysis was performed using "SYNCWUS", a computer program developed by the University of Virginia/3/.

An assumed unbalance mass of 4.3×10^{-2} N-m (6.1 oz-in) at a phase angle of 0° was placed at the centre of the rotor. This was done to ensure that the first mode would be excited. The unbalance mass M_u was determined from the following equation:

$$M_u = \frac{4W}{N}$$

where W = Weight of the rotor (lb)
N = Operating speed (rpm)
M_u = Unbalance mass (oz-in)

For a rotor weight of 2494.8 kg (5500 lbs) and an operating speed of approximately 3600 rpm, the calculated mass unbalance is

$$M_u = \frac{4(5500 \text{ lb})}{3600 \text{ rpm}} = 6.1 \text{ oz-in } (4.3 \times 10^{-2} \text{ N-m})$$

The results of the rotor unbalance response analyses for journal bearing diametral clearances of 152.4 µm (0.006 in) to 330.2 µm (0.013 in) are presented in Figures 15-38. The X and Y displacements as well as phase are plotted for rotor stations No 3 (outboard bearing), No 17 (centre of rotor), and No 31 (inboard bearing). These amplitudes should only be used for comparison purposes since they are dependent on the assumed amount and location of the unbalance.

Table 5 lists the rotor first natural frequencies, indicated by the peak response displacements, for the range of bearing diametral clearances.

Table 6 shows the rotor displacement levels at 3600 rpm, at the three stations (3, 17 and 31), for the range for bearing diametral clearances.

These results indicate that the rotor response is quite sensitive to small increases in the journal bearing diametral clearances, particularly in the horizontal (X) direction.

At the typical journal bearing diametral clearance of 0.008" (as measured in the field), the rotor first critical speed is approximately 3000 rpm (50 Hz) in the horizontal direction.

This frequency compares well with that determined experimentally (53.5 Hz), as discussed in Section 3.2.

5.0 ROTOR CRITICAL SPEED MAP ANALYSIS

Figure 41 is a critical speed map for the motor. The map is a plot of the first three undamped critical speeds versus a range of selected bearing stiffness values. The critical speeds increase as the stiffness of the bearings is increased. However, these curves all become asymptotic to an upper limiting speed. In this region, the critical speeds are insensitive to increasing bearing stiffness and the bearing damping is no longer effective for the suppression of vibration. The rotor as modelled in Figure 13 was used for the analysis. The University of Virginia computer program "CRTSP2"/4/ was used for the analysis.

For the case of the journal bearing with diametral clearance of 203.2 μm (0.008 in), the bearing stiffnesses for a range of speeds were determined using "FINGBRG1" computer program. These results are presented in Table 7.

Using the support stiffnesses determined experimentally, the combined bearing-support stiffnesses were estimated as follows:

$$K_{EX} = \frac{K_{XX} \cdot K_{SH}}{K_{XX} + K_{SH}}$$

$$K_{EY} = \frac{K_{YY} \cdot K_{SV}}{K_{YY} + K_{SV}}$$

where K_{XX} = orthogonal bearing stiffness in the horizontal (X) direction

K_{YY} = orthogonal bearing stiffness in the vertical (Y) direction

K_{SH} = support stiffness in the horizontal direction

K_{SV} = support stiffness in the vertical direction

K_{EX} = bearing-support equivalent stiffness in the horizontal (X) direction

K_{EY} = bearing-support equivalent stiffness in the vertical (Y) direction

These equivalent (bearing-support) stiffnesses are tabulated in Table 8 for a range of rotor speeds, and plotted on the critical speed map as shown in Figure 41.

The intersection of the bearing-support stiffness curve and the rotor critical speed curves identify rotor critical speeds. Figure 41 shows that the rotor critical speeds (first mode), for the case of 0.008" diametral clearance, are at approximately 2950 rpm and 2750 rpm for the horizontal (X) and vertical (Y) directions, respectively. The rotor first critical speeds, obtained from the unbalance response analysis (see section 4.3) are 3000 rpm and 3800 rpm for the horizontal and vertical directions, respectively. Differences in rotor critical speeds as determined by the two methods can be attributed to the estimates of the equivalent stiffnesses (K_{EX}, K_{EY}) used in producing the rotor critical speed map.

In the unbalance response analysis, the effects of the bearing cross-coupled stiffnesses and damping are considered. From Table 7, it can be observed that in some cases the cross coupled stiffnesses (K_{XY}, K_{YX}) are significantly larger than the orthogonal stiffnesses (K_{XX},K_{YY}) and as such would tend to alter the rotor first critical speeds. The rotor first critical speeds obtained from the unbalance response analysis are more accurate because the model used is more representative of the rotor-bearing-support system.

The critical speed map provides insight into the relationship between the stiffnesses of the shaft and rotor support system and their effects on the critical speeds and therefore provides an overview of the type of practical changes that would be required to effectively detune rotor critical speed problems.

A typical rotor critical speed map for a first mode is shown in Figure 42. There are three general regions of interest. They are:

Region No 1 In this region, the shaft is considered rigid when compared with the bearing-support system. The rotor natural frequency is very sensitive to the rotor bearing-support stiffness. Thus the rotor natural frequency can be shifted easily by increasing or decreasing the rotor bearing-support stiffness by a small amount.

Region No 2 In this region, the shaft is considered semi-rigid when compared with the bearing-support system. A significant bearing-support stiffness change would be required for an appreciable change in rotor natural frequency.

Region No 3 In this region, the shaft is considered to be flexible when compared with the rigid bearings. In this region, the rotor natural frequency has very low sensitivity to the rotor bearing-support stiffness. Appreciable changes in rotor natural frequency can only be attained with a redesign of the shaft system.

For the Bruce 'B' NGS Unit 5 - Boiler Feed Pump Motor #1, the operating speed is close to the borderline between Region 2 and Region 3. The unbalance response analysis has shown that the motor (typical bearing clearance of 0.008") is operating close to the first rotor critical speed. Mode shapes obtained by the unbalance response analysis for the rotor natural frequency in the horizontal direction (ie, 3000 rpm) and in the vertical direction (ie, 3800 rpm) are shown in Figures 39 and 40. These mode shapes illustrate the significant degree of bending in the shaft and confirm that the system has characteristics typical of those found in Region 2.

The results of the theoretical rotor dynamics analysis indicate that the rotor critical speeds can likely be shifted by altering the bearing clearance to reduce the bearing oil-film stiffness and/or softening the bearing support. The analysis also indicates the manufacturer's specification which states

> "The first critical speed in percentage of full load speed is to be greater than 125%"/1/.

most probably could not be attained with the existing design. The critical speed map for the first mode (Figure 41) shows that the bearing oil film-support stiffness combination must be 7.88 x 10^8 N/m (4.5 x 10^6 lb/in) for the first rotor natural frequency to be 4500 rpm (ie. 1.25 x 3600 rpm). With the existing rotor size and weight as well as support configuration, it would be difficult to produce this required stiffness.

6.0 CONCLUSIONS

6.1 Motor vibrations are considered high and corrective actions should be taken.

6.2 Experimental and theoretical rotor dynamics analyses indicate that there is a rotor critical speed near the operating speed which is most likely the main cause of the amplified vibration.

7.0 RECOMMENDATION

7.1 An investigation should be carried out to determine practical bearing and/or structural modifications that can be implemented to detune the rotor-bearing-support system and thereby significantly reduce rotor amplification.

Approved by:

G.J. Clarke
Manager
Mechanical Research Dept

Submitted by:

F. Perricone
Unit Head, Structural Dynamics
Plant Equipment Dynamics Section

N.A. Jagan
Engineer
Plant Equipment Dynamics Section

NAJ/FP:am/lg

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3. Salamone, D.J., Gunter, E.J., "Synchronous Unbalance Response of a Multimass Flexible Rotor Considering Shaft Warp and Disk Skew", University of Virginia, Report No UVA/464761/ME77/142, May 1977.

4. Li, D.F., Gunter, E.J., Allaire, P.E., "Undamped Critical Speed Analysis of Dual-level Rotor Systems - A Manual for Use with Computer Program CRTSP2", University of Virginia, Report No UVA/464761/ME76/136.

TABLE 1

INSTRUMENTATION LIST

1.	Accelerometer	Bruel & Kjaer Model 4368
2.	Charge Amplifier	Bruel & Kjaer Model 2635
3.	Displacement Transducer	Bently Nevada Model 7200 Series
4.	Displacement Transducer Amplifier	Ontario Hydro Research
5.	Phase Transducer	Electro Corporation Model 3030AN
6.	Digital Vector Filter	Bently Nevada Model DVF2
7.	Dual Channel FFT Analyzer	Ono Sokki CF920
8.	Digital Plotter	Hewlett Packard 7470A
9.	Oscilloscope	Textronic Model 5110 MOD 709V
10.	Tape Recorder	Honeywell Model 101
11.	Filter	KronHite Model 3343
12.	Vibration Calibrator	General Radio Model 1557-A
13.	Hydraulic Shaker	Zonic Model 1104 MOD 2
14.	Absolute Rotor Displacement Instrumentation	Ontario Hydro Research

TABLE 2

MOTOR ROTOR OVERALL DISPLACEMENT LEVELS

Measurement Location	Displacement µm p-p	Figure No
OB VERT	50-55	1
OB HOR	75-77	2
IB VERT	48-54	3
IB HOR	52-60	4

TABLE 3

MOTOR CASING OVERALL VIBRATION LEVELS

Measurement Location	Velocity mm/s	Displacement µm p-p	Figure No
OB VERT	4.7-5.3	47-80	5
OB HOR	5.9-6.2	50-85	6
OB AXIAL	7.7-8.5	70-100	7
IB VERT	7.2-8.0	50-83	8
IB HOR	4.7-5.1	68-99	9

TABLE 4

JOURNAL BEARING STIFFNESS AND DAMPING COEFFICIENTS AT 3600 RPM

Diametral Clearance μm (in)	K_{XX} N/m (lb/in)	K_{XY} N/m (lb/in)	K_{YX} N/m (lb/in)	K_{YY} N/m (lb/in)	C_{XX} N·s/m (lb·s/in)	C_{XY} N·s/m (lb·s/in)	C_{YX} N·s/m (lb·s/in)	C_{YY} N·s/m (lb·s/in)
152.4 (0.006)	3.31×10^{8} (1.89×10^{6})	6.70×10^{8} (3.83×10^{6})	-8.28×10^{8} (-4.73×10^{6})	1.96×10^{8} (1.12×10^{6})	33.38×10^{5} (19.06×10^{3})	-7.99×10^{5} (-4.56×10^{3})	-12.04×10^{5} (-6.88×10^{3})	42.49×10^{5} (24.26×10^{3})
177.8 (0.007)	2.81×10^{8} (1.60×10^{6})	4.24×10^{8} (2.42×10^{6})	-5.97×10^{8} (-3.41×10^{6})	1.83×10^{8} (1.04×10^{6})	22.66×10^{5} (12.94×10^{3})	-6.99×10^{5} (-3.99×10^{3})	-10.07×10^{5} (-5.75×10^{3})	30.08×10^{5} (17.17×10^{3})
203.2 (0.008)	2.44×10^{8} (1.39×10^{6})	2.84×10^{8} (1.62×10^{6})	-4.68×10^{8} (-2.68×10^{6})	1.77×10^{8} (1.01×10^{6})	16.49×10^{5} (9.42×10^{3})	-6.25×10^{5} (-3.57×10^{3})	-8.75×10^{5} (-4.99×10^{3})	23.18×10^{5} (13.24×10^{3})
228.6 (0.009)	2.15×10^{8} (1.23×10^{6})	1.98×10^{8} (1.13×10^{6})	-3.91×10^{8} (-2.23×10^{6})	1.75×10^{8} (0.997×10^{6})	12.64×10^{5} (7.22×10^{3})	-5.67×10^{5} (-3.24×10^{3})	-7.79×10^{5} (-4.45×10^{3})	19.04×10^{5} (10.87×10^{3})
254.0 (0.010)	1.92×10^{8} (1.09×10^{6})	1.42×10^{8} (0.81×10^{6})	-3.40×10^{8} (-1.94×10^{6})	1.75×10^{8} (1.00×10^{6})	10.07×10^{5} (5.75×10^{3})	-5.21×10^{5} (-2.97×10^{3})	-7.07×10^{5} (-4.04×10^{3})	16.37×10^{5} (9.35×10^{3})
279.4 (0.011)	1.73×10^{8} (0.99×10^{6})	1.03×10^{8} (0.59×10^{6})	-3.05×10^{8} (-1.74×10^{6})	1.77×10^{8} (1.01×10^{6})	8.26×10^{5} (4.72×10^{3})	-4.82×10^{5} (-2.75×10^{3})	-6.48×10^{5} (-3.70×10^{3})	14.57×10^{5} (8.32×10^{3})
304.8 (0.012)	1.58×10^{8} (0.90×10^{6})	0.75×10^{8} (0.43×10^{6})	-2.80×10^{8} (-1.60×10^{6})	1.80×10^{8} (1.03×10^{6})	6.95×10^{5} (3.97×10^{3})	-4.50×10^{5} (-2.57×10^{3})	-6.02×10^{5} (-3.44×10^{3})	13.27×10^{5} (7.58×10^{3})
330.2 (0.013)	1.45×10^{8} (0.83×10^{6})	0.56×10^{8} (0.32×10^{6})	-2.61×10^{8} (-1.49×10^{6})	1.84×10^{8} (1.05×10^{6})	5.95×10^{5} (3.40×10^{3})	-4.22×10^{5} (-2.41×10^{3})	-5.62×10^{5} (-3.21×10^{3})	12.31×10^{5} (7.03×10^{3})

TABLE 5

ROTOR FIRST CRITICAL SPEEDS VERSUS BEARING DIAMETRAL CLEARANCE

Bearing Diametral Clearance	Rotor First Critical Speed (rpm)					
	Station No 3 Outboard Bearing		Station No 17 Centre of Rotor		Station No 31 Inboard Bearing	
	X	Y	X	Y	X	Y
152.4 µm (0.006 in)	3400	3800	3400	3800	3400	3800
177.8 µm (0.007 in)	3200	3800	3200	3800	3200	3800
203.2 µm (0.008 in)	3000	3800	2800	3800	3000	3800
228.6 µm (0.009 in)	2600	2800	2600	2800	2600	2800
254.0 µm (0.010 in)	2600	2600	2600	2600	2600	2600
279.4 µm (0.011 in)	2400	2400	2400	2400	2400	2400
304.8 µm (0.012 in)	2400	2400	2400	2400	2400	2400
330.2 µm (0.013 in)	2400	2400	2400	2400	2400	2400

TABLE 6

ROTOR DISPLACEMENT LEVELS AT 3600 RPM VERSUS BEARING CLEARANCE

		BEARING DIAMETRAL CLEARANCE							
		152.4 μm (0.006 in)	177.8 μm (0.007 in)	203.2 μm (0.008 in)	228.6 μm (0.009 in)	254.0 μm (0.010 in)	279.4 μm (0.011 in)	304.8 μm (0.012 in)	330.2 μm (0.013 in)
Rotor OB Brg (Station #3) Displacement (μm p-p)	X-dir	5.03	4.32	3.72	3.31	3.06	2.92	2.86	2.83
	Y-dir	14.59	12.97	11.77	11.0	10.50	10.17	9.93	9.76
Rotor Centre (Station #17) Displacement (μm p-p)	X-dir	7.70	5.82	4.48	3.64	3.16	2.90	2.75	2.67
	Y-dir	23.42	20.11	17.75	16.24	15.29	14.66	14.20	13.91
Rotor IB Brg (Station #31) Displacement (μm p-p)	X-dir	5.09	4.39	3.80	3.40	3.15	3.01	2.94	2.92
	Y-dir	14.78	13.17	12.0	11.25	10.78	10.49	10.29	10.16

TABLE 7

JOURNAL BEARING STIFFNESS COEFFICIENTS VERSUS ROTOR SPEED FOR BEARING DIAMETRAL CLEARANCE OF 203.2 μm (0.008 in)

Rotor Speed rpm	K_{XX} N/m (lb/in)	K_{XY} N/m (lb/in)	K_{YX} N/m (lb/in)	K_{YY} N/m (lb/in)
900	2.33×10^8 (1.33×10^6)	0.27×10^8 (0.16×10^6)	-4.54×10^8 (-2.59×10^6)	4.18×10^8 (2.39×10^6)
1800	2.38×10^8 (1.36×10^6)	1.24×10^8 (0.71×10^6)	-4.21×10^8 (-2.41×10^6)	2.63×10^8 (1.50×10^6)
2700	2.42×10^8 (1.38×10^6)	2.05×10^8 (1.17×10^6)	-4.35×10^8 (-2.48×10^6)	2.05×10^8 (1.17×10^6)
3600	2.44×10^8 (1.39×10^6)	2.84×10^8 (1.62×10^6)	-4.68×10^8 (-2.68×10^6)	1.77×10^8 (1.01×10^6)
4500	2.45×10^8 (1.40×10^6)	3.62×10^8 (2.06×10^6)	-5.15×10^8 (-2.94×10^6)	1.60×10^8 (0.92×10^6)

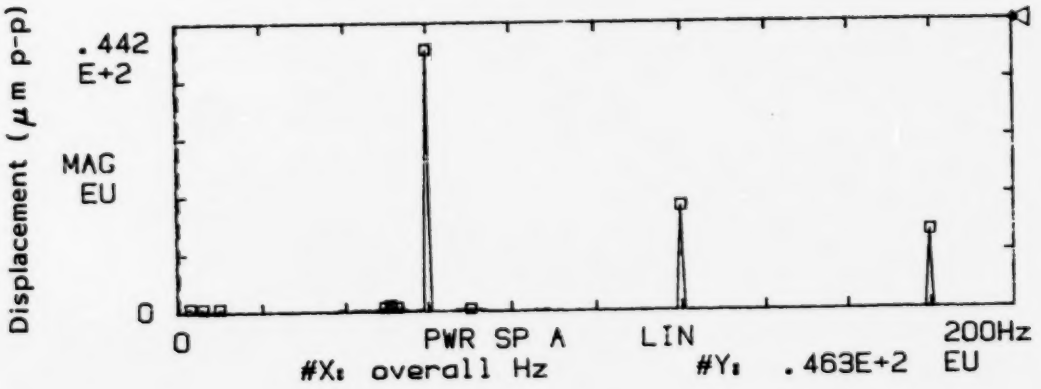
TABLE 8

BEARING-SUPPORT STIFFNESS VERSUS ROTOR SPEED
- BEARING DIAMETRAL CLEARANCE = 203.2 μm (0.008 in)

Rotor Speed rpm	K_{EX} N/m (lb/in)	K_{EY} N/m (lb/in)
900	1.58×10^8 (0.903×10^6)	1.94×10^8 (1.11×10^6)
1800	1.60×10^8 (0.916×10^6)	1.52×10^8 (0.87×10^6)
2700	1.62×10^8 (0.925×10^6)	1.31×10^8 (0.748×10^6)
3600	1.63×10^8 (0.930×10^6)	1.19×10^8 (0.678×10^6)
4500	1.63×10^8 (0.932×10^6)	1.11×10^8 (0.635×10^6)

TEST#4 BRUCE NGSB U#5 BFP MTR#1 OB ROTOR VERT DISPL
200Hz A: AC/ 2V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	60.00Hz	.403E+2	EU
2	120.00	.161E+2	
3	180.00	.121E+2	
4	51.02	.109E+1	
5	2.91	.748E+0	
6	49.48	.817E+0	
7	52.55	.727E+0	
8	6.00	.609E+0	
9	9.97	.556E+0	
10	70.41	.508E+0	



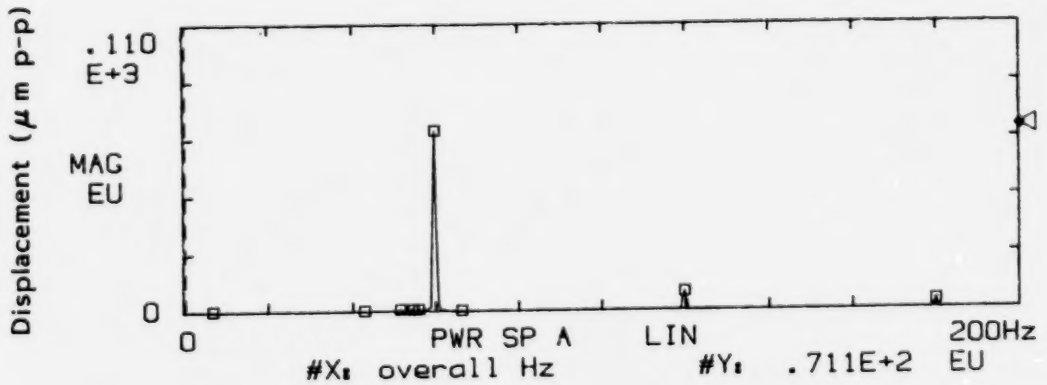
03:54

FIGURE 1

MOTOR OUTBOARD BEARING ROTOR DISPLACEMENT VERTICAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 OB ROTOR HOR DISPL
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	60.00Hz	.701E+2	EU
2	120.02	.740E+1	
3	180.02	.319E+1	
4	51.55	.742E+0	
5	54.47	.771E+0	
6	55.97	.692E+0	
7	6.95	.409E+0	
8	42.94	.354E+0	
9	66.50	.367E+0	
10			



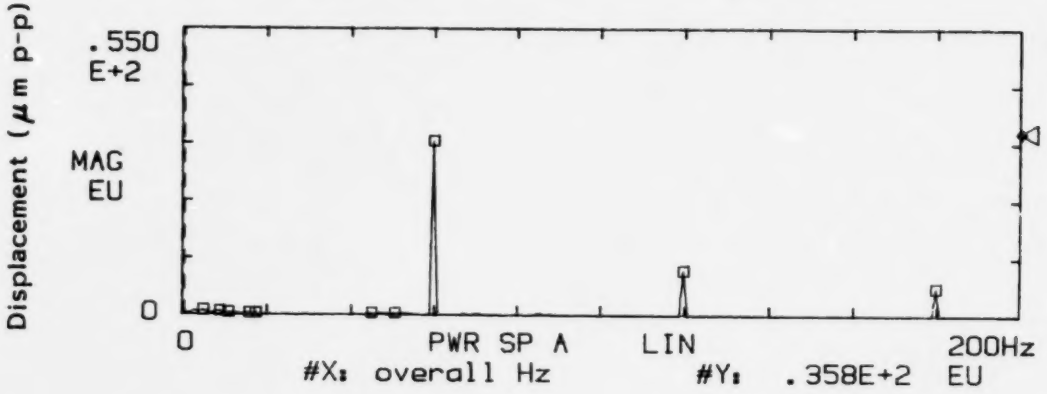
04:00

FIGURE 2

MOTOR OUTBOARD BEARING ROTOR DISPLACEMENT HORIZONTAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 IB ROTOR VERT DISPL
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	60.00Hz	.338E+2	EU
2	120.02	.880E+1	
3	180.02	.538E+1	
4	5.02	.861E+0	
5	8.95	.699E+0	
6	10.94	.534E+0	
7	15.95	.424E+0	
8	17.52	.367E+0	
9	44.94	.423E+0	
10	50.45	.348E+0	

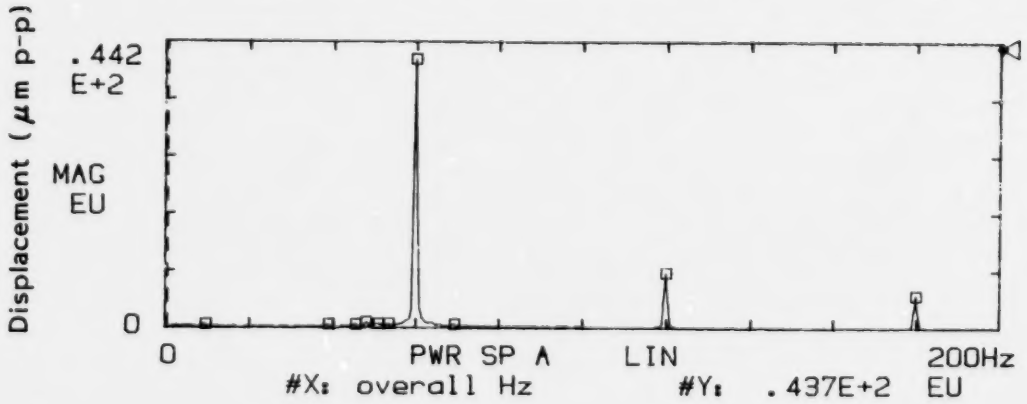


04:03

FIGURE 3
MOTOR INBOARD BEARING ROTOR DISPLACEMENT VERTICAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 IB ROTOR HOR DISPL
200Hz A: AC/ 2V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	60.00Hz	.416E+2	EU
2	120.00	.869E+1	
3	180.00	.521E+1	
4	48.02	.845E+0	
5	50.95	.802E+0	
6	53.47	.693E+0	
7	9.48	.591E+0	
8	38.97	.564E+0	
9	45.48	.569E+0	
10	69.47	.586E+0	



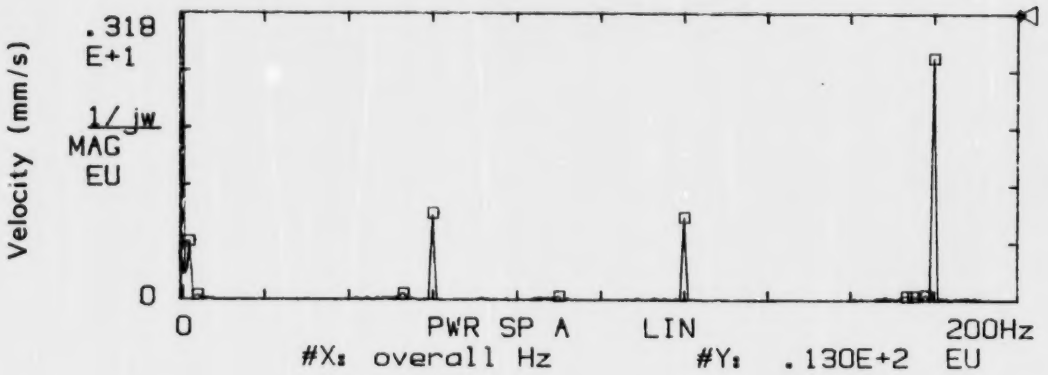
04:06

FIGURE 4

MOTOR INBOARD BEARING ROTOR DISPLACEMENT HORIZONTAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 OB STRUCT VERT VELOCITY
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	180.00Hz	.271E+1	EU
2	60.00	.962E+0	
3	120.00	.931E+0	
4	2.00	.657E+0	
5	53.00	.727E-1	
6	178.00	.684E-1	
7	4.00	.556E-1	
8	173.50	.500E-1	
9	175.00	.545E-1	
10	89.98	.415E-1	

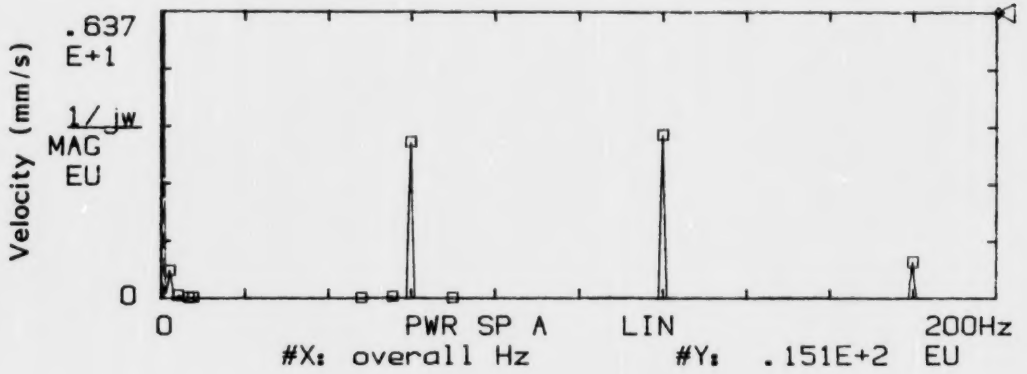


02: 46

FIGURE 5
MOTOR OUTBOARD BEARING CASING VELOCITY VERTICAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 OB STRUCT HOR VELOCITY
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	120.00Hz	.364E+1	EU
2	60.00	.347E+1	
3	180.00	.819E+0	
4	2.00	.633E+0	
5	4.02	.696E-1	
6	55.50	.416E-1	
7	6.48	.272E-1	
8	7.50	.233E-1	
9	48.00	.212E-1	
10	70.00	.203E-1	

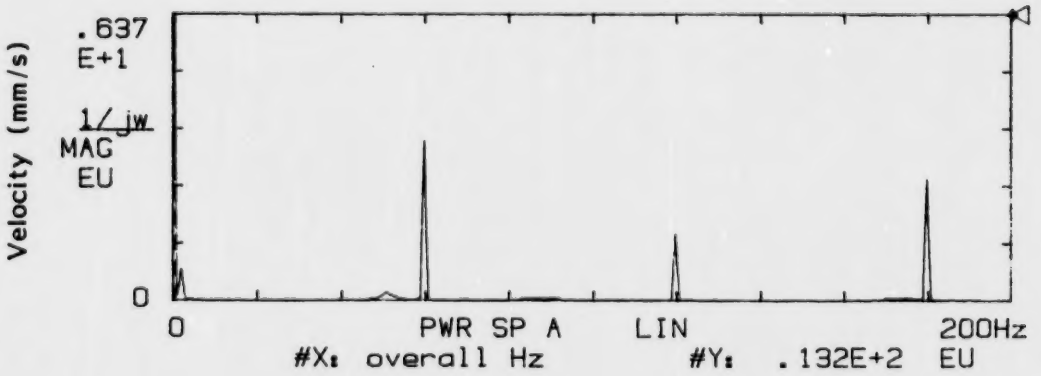


03: 21

FIGURE 6
MOTOR OUTBOARD CASING VELOCITY HORIZONTAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 OB STRUCT AXIA VELOCITY
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	60.00Hz	.357E+1	EU
2	180.00	.272E+1	
3	120.00	.148E+1	
4	2.00	.711E+0	
5	51.00	.182E+0	
6	4.00	.676E-1	
7	54.50	.597E-1	
8	85.50	.732E-1	
9	87.50	.656E-1	
10	90.50	.621E-1	

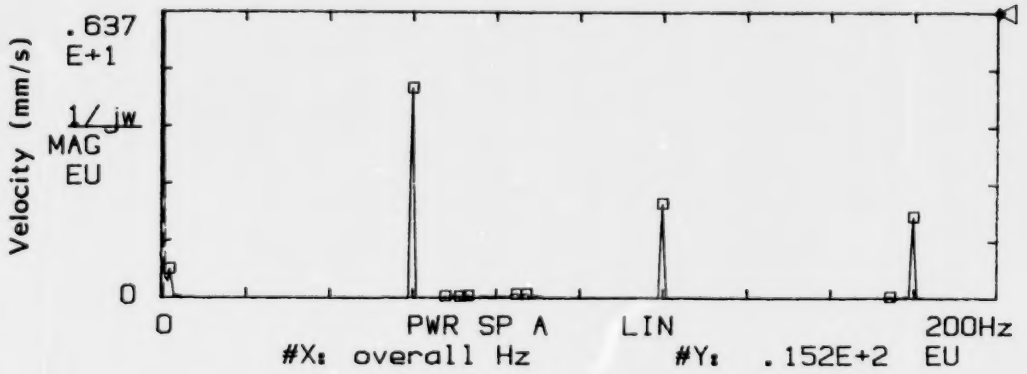


03: 30

FIGURE 7
MOTOR OUTBOARD BEARING CASING VELOCITY AXIAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 IB STRUCT VERT VELOCITY
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

	PWR SPECTRUM	ChA	
1	60.00Hz	.472E+1	EU
2	120.00	.211E+1	
3	180.00	.184E+1	
4	2.00	.652E+0	
5	85.00	.108E+0	
6	87.50	.106E+0	
7	73.50	.598E-1	
8	68.00	.405E-1	
9	71.50	.420E-1	
10	174.50	.491E-1	



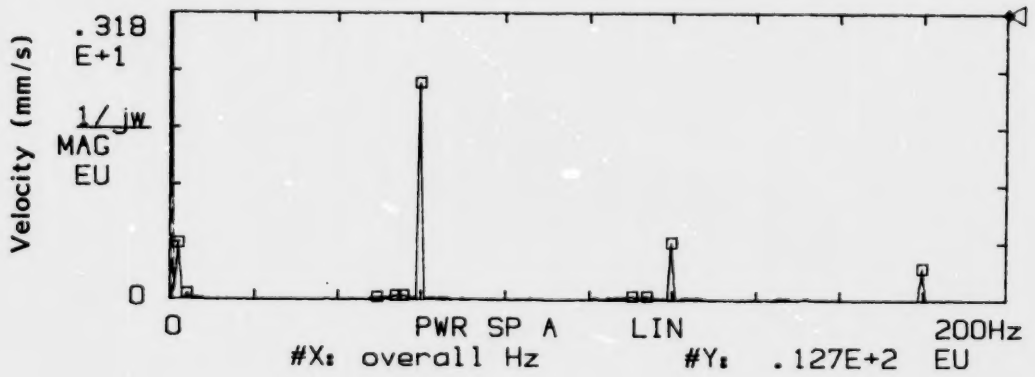
03: 24

FIGURE 8

MOTOR INBOARD BEARING CASING VELOCITY VERTICAL DIRECTION

TEST#4 BRUCE NGSB U#5 BFP MTR#1 IB STRUCT HOR VELOCITY
200Hz A: AC/ 1V B: AC/ 2V S. SUM 16/16 DUAL 1k

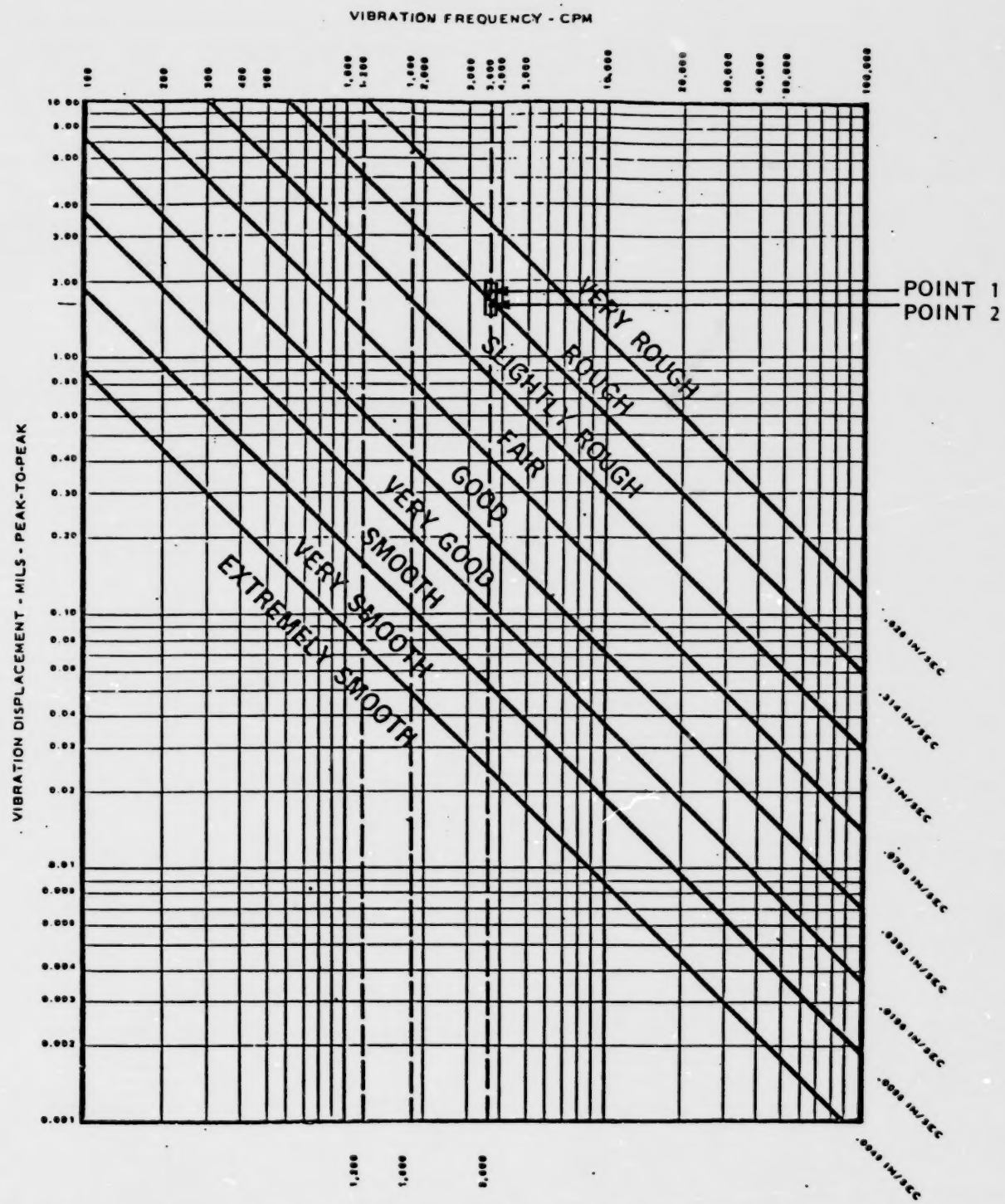
	PWR SPECTRUM	ChA	
1	60.00Hz	.242E+1	EU
2	120.00	.639E+0	
3	2.00	.637E+0	
4	180.00	.366E+0	
5	4.02	.707E-1	
6	54.00	.529E-1	
7	56.00	.572E-1	
8	110.48	.420E-1	
9	114.00	.444E-1	
10	49.50	.305E-1	



03: 27

FIGURE 9

MOTOR INBOARD BEARING CASING VELOCITY
HORIZONTAL DIRECTION



Point 1 - Maximum Velocity Level Measured on Motor Outboard Bearing Casing
Point 2 - Maximum Velocity Level Measured on Motor Inboard Bearing Casing

FIGURE 10

MAXIMUM MOTOR VELOCITY LEVELS PLOTTED ON A GENERAL MACHINERY VIBRATION SEVERITY CHART

175168-RD

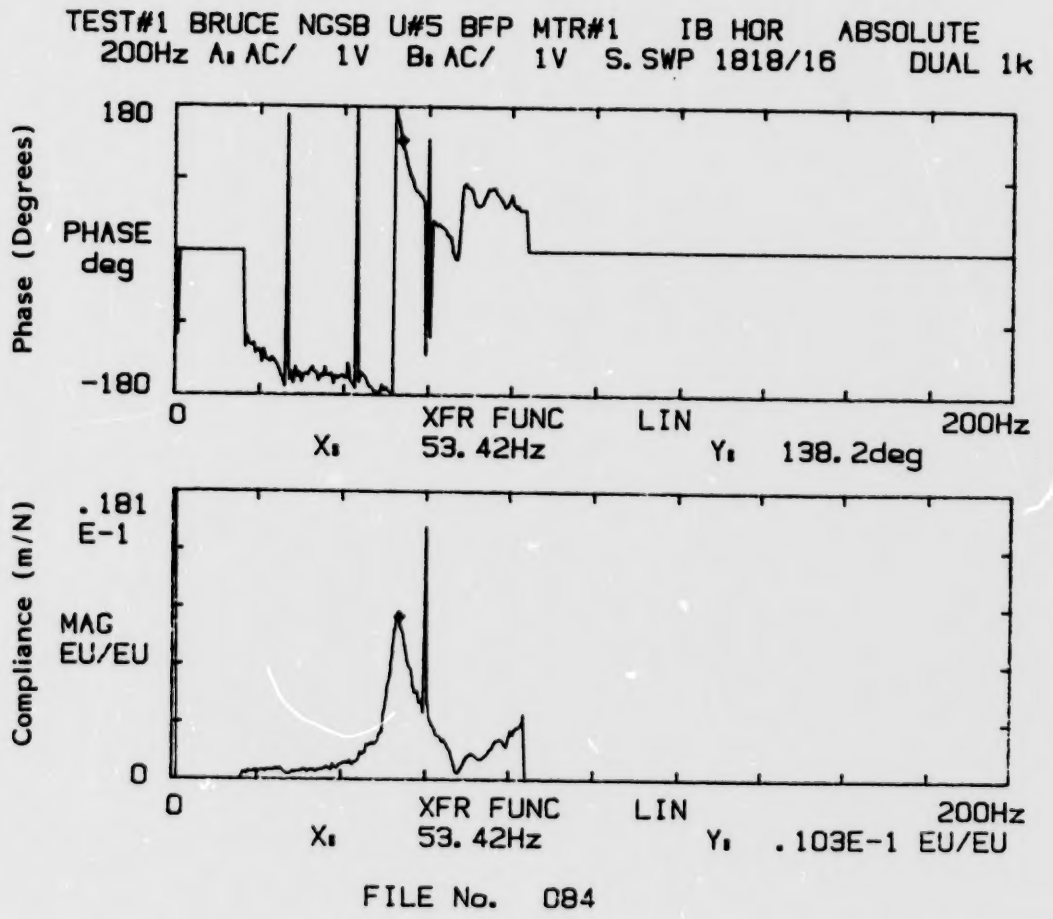


FIGURE 11

MOTOR INBOARD BEARING ROTOR ABSOLUTE COMPLIANCE TRANSFER FUNCTION AND PHASE (HORIZONTAL DIRECTION)

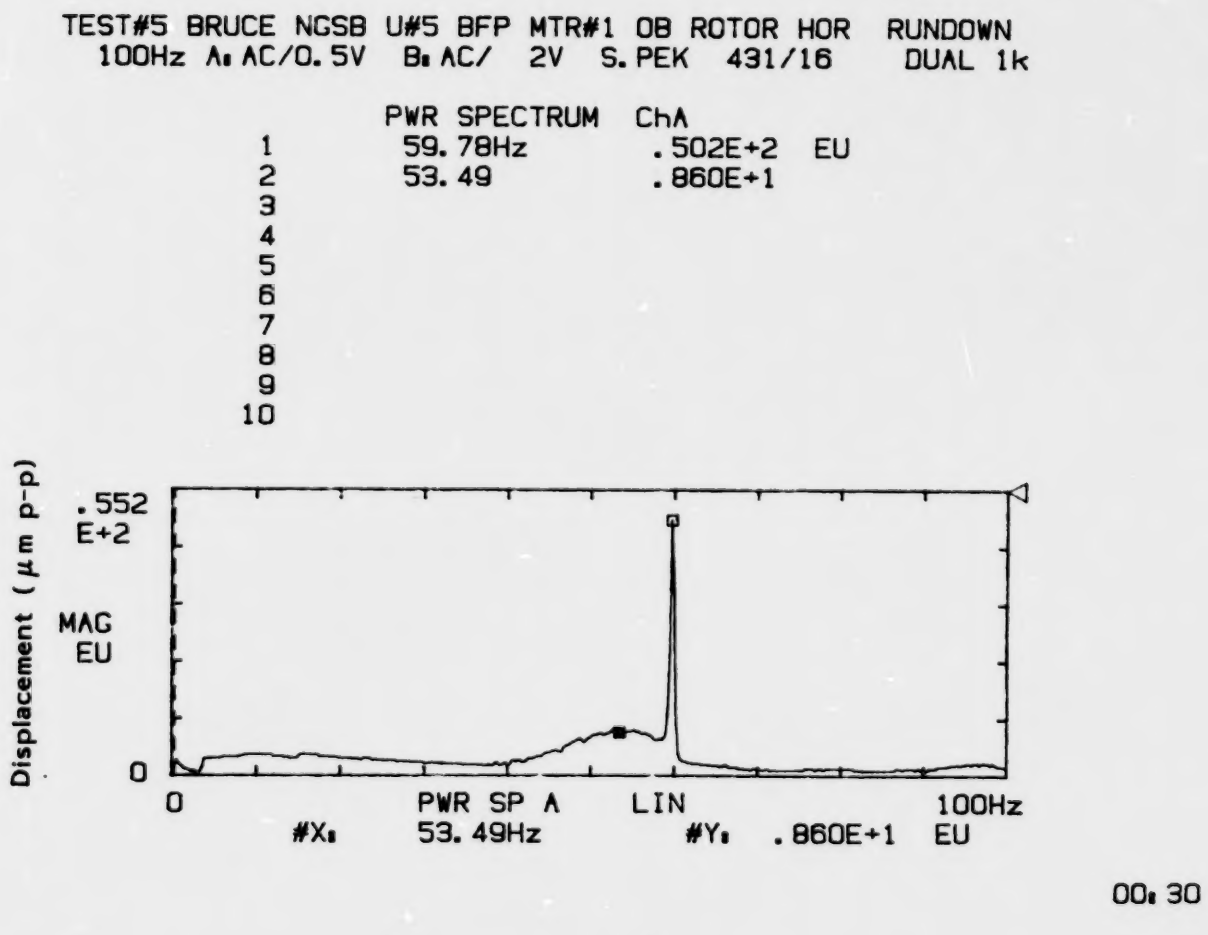
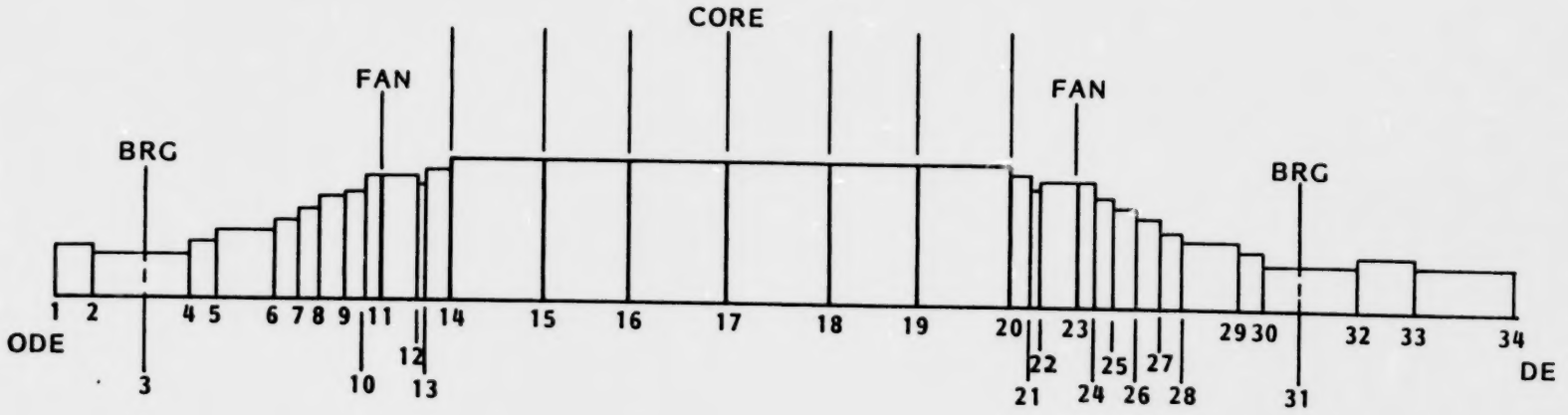


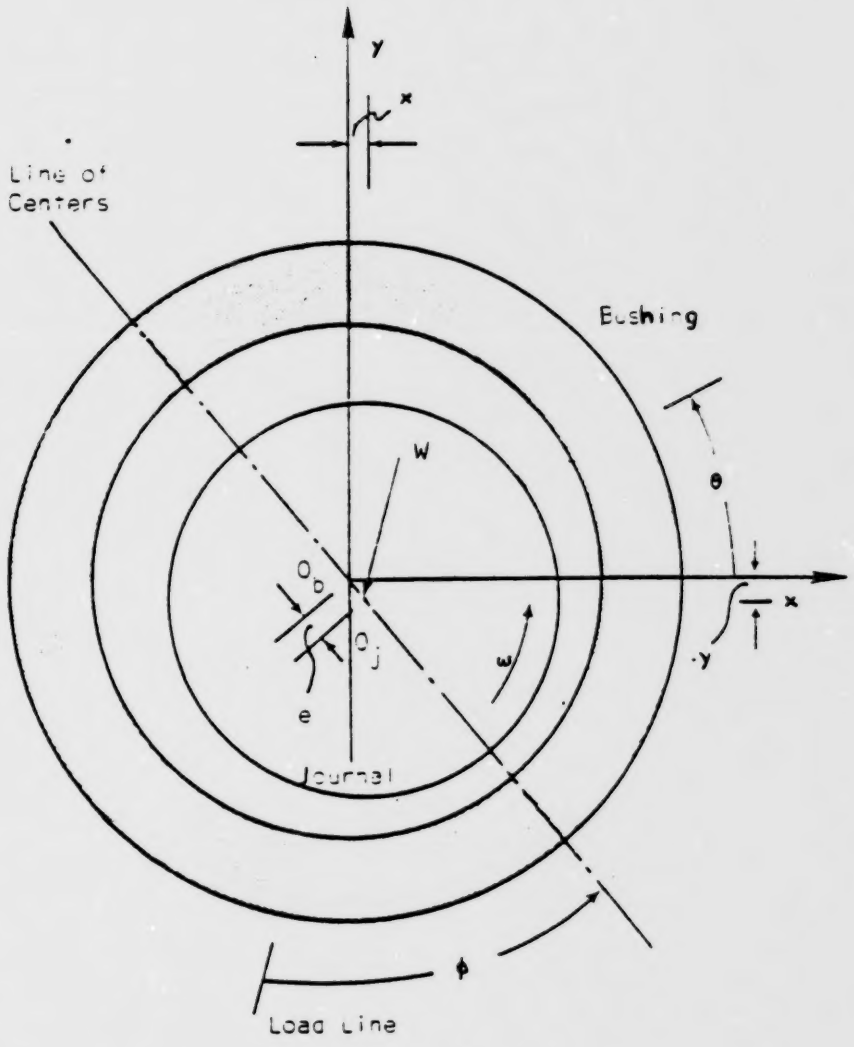
FIGURE 12

MOTOR OUTBOARD BEARING ROTOR DISPLACEMENT
PEAK HOLD RUNDOWN SPECTRUM (HORIZONTAL DIRECTION)



MASS STATION	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
DIAMETER OF SHAFT mm SECTION TO THE RIGHT OF MASS STATION (in)	157.5 (6.2)	139.7 (5.5)	139.7 (5.5)	165.1 (6.5)	177.8 (7)	202.6 (7.98)	252.2 (9.93)	301.8 (11.88)	342.9 (13.5)	365.8 (14.4)	365.8 (14.4)	355.6 (14)	367.8 (14.48)	368.3 (14.5)	368.3 (14.5)	368.3 (14.5)	368.3 (14.5)
LENGTH OF SHAFT mm SECTION TO THE RIGHT OF MASS STATION (in)	48.3 (1.9)	92.7 (3.65)	59.7 (2.35)	45.7 (1.8)	124.5 (4.9)	58.4 (2.3)	58.4 (2.3)	58.4 (2.3)	40.6 (1.6)	30.5 (1.2)	76.2 (3.0)	12.7 (0.5)	33.0 (1.3)	128.7 (5.07)	128.7 (5.07)	128.7 (5.07)	128.7 (5.07)
ADDITIONAL WEIGHT kg (lb)	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -	22.7 (50)	- -	- -	100.2 (220.8)	200.4 (441.7)	200.4 (441.7)	200.4 (441.7)
MASS STATION	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
DIAMETER OF SHAFT mm SECTION TO THE RIGHT OF MASS STATION (in)	368.3 (14.5)	368.3 (14.5)	367.8 (14.48)	355.6 (14)	365.8 (14.4)	365.8 (14.4)	342.9 (13.5)	301.8 (11.88)	252.2 (9.93)	202.6 (7.98)	177.8 (7)	165.1 (6.5)	139.7 (5.5)	139.7 (5.5)	157.5 (6.2)	139.7 (5.5)	 (0)
LENGTH OF SHAFT mm SECTION TO THE RIGHT OF MASS STATION (in)	128.7 (5.07)	128.7 (5.07)	33.0 (1.3)	12.7 (0.5)	76.2 (3.0)	30.5 (1.2)	40.6 (1.6)	58.4 (2.3)	58.4 (2.3)	58.4 (2.3)	124.5 (4.9)	45.7 (1.8)	59.7 (2.35)	92.7 (3.65)	154.9 (6.1)	236.2 (9.3)	 (0)
ADDITIONAL WEIGHT kg (lb)	200.4 (441.7)	200.4 (441.7)	100.2 (220.8)	- -	- -	22.7 (50)	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -	- -

Figure 13
The Rotor Model



SIGN CONVENTION FOR BEARING ANALYSIS

FIGURE 14

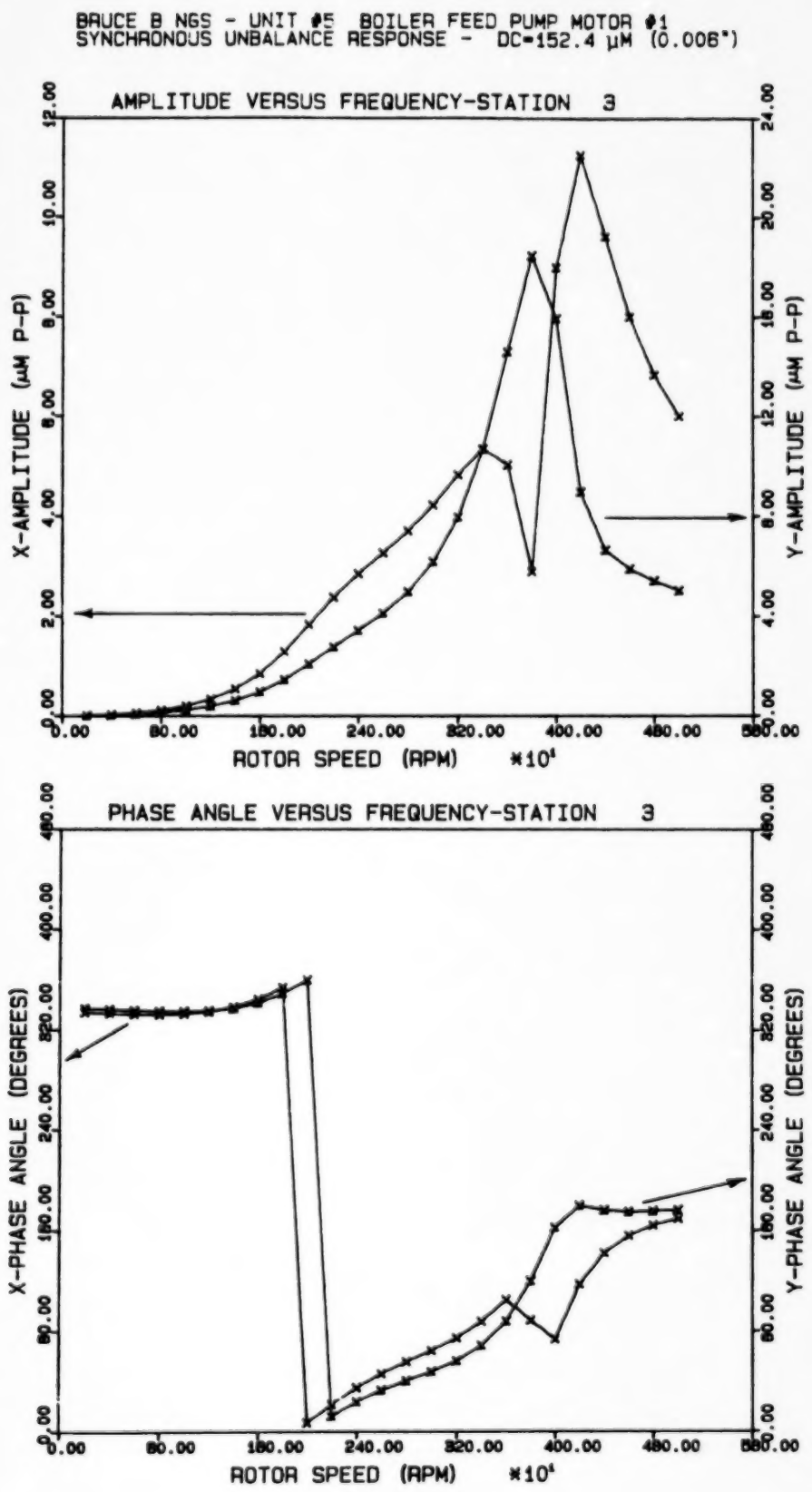

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=152.4 µM (0.006")
AMPLITUDE VERSUS FREQUENCY-STATION 3
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 3
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10¹

Figure 15

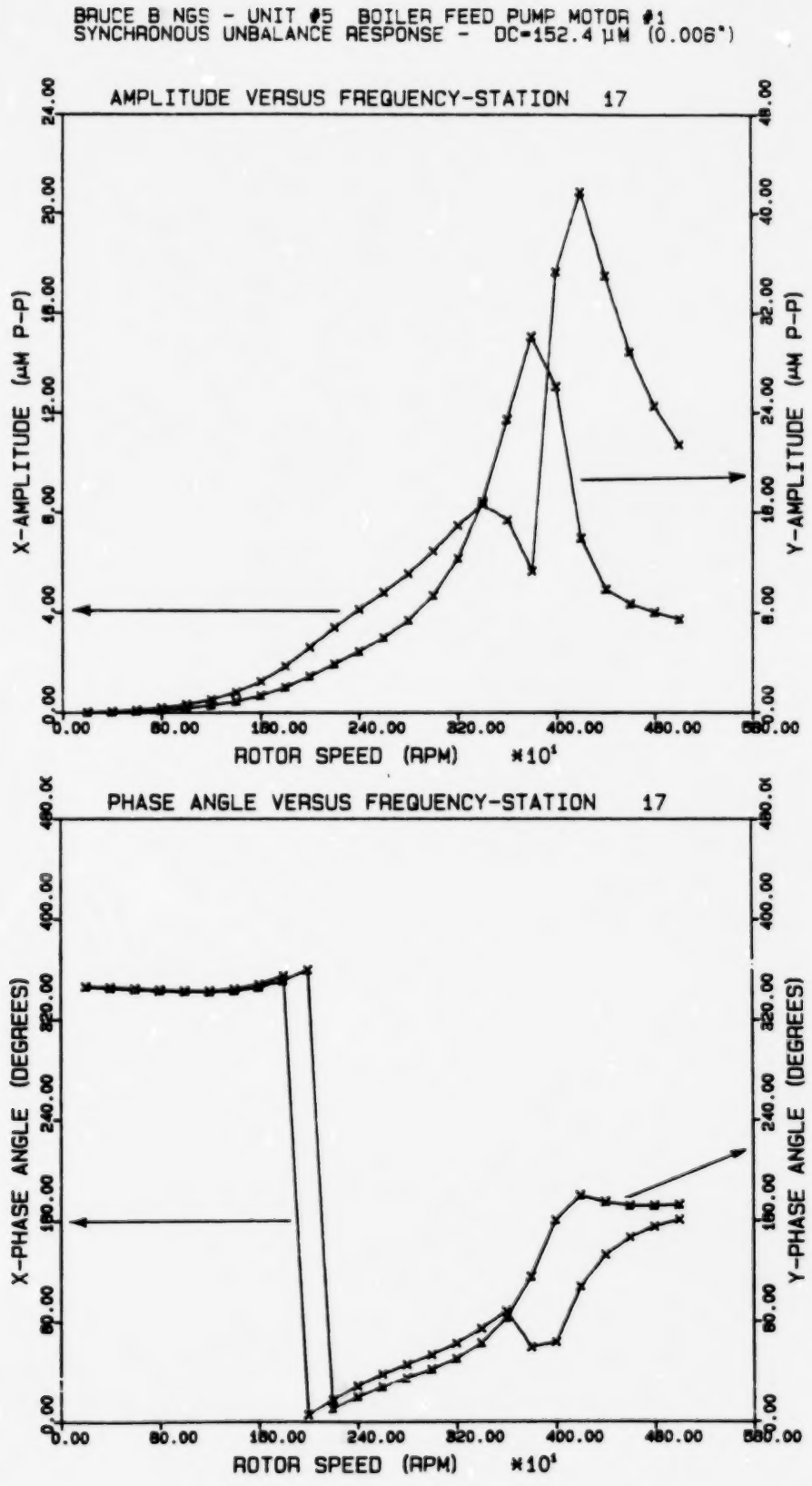

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=152.4 µM (0.006")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10¹

Figure 16

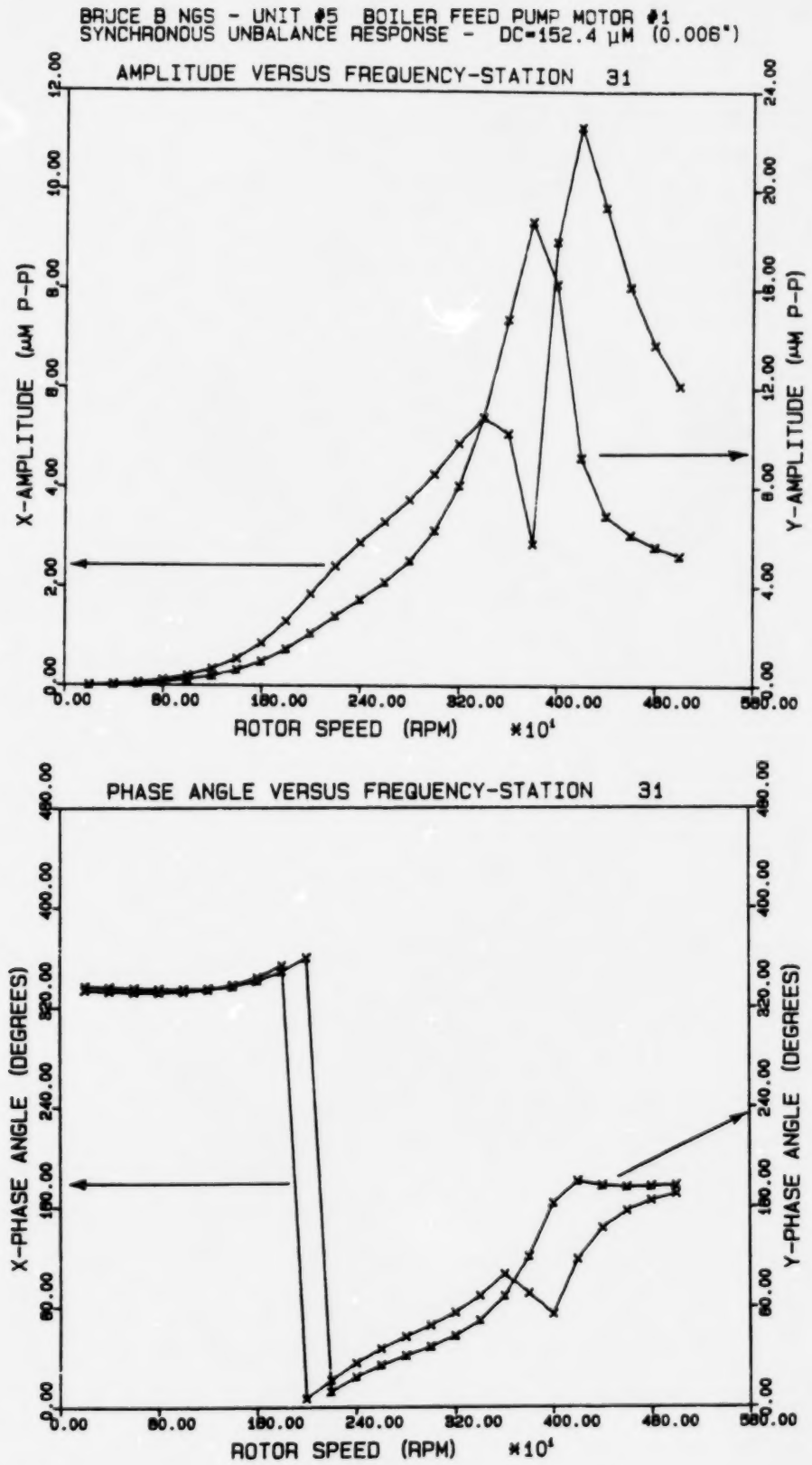

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=152.4 µM (0.006")
AMPLITUDE VERSUS FREQUENCY-STATION 31
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 31
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10¹

Figure 17

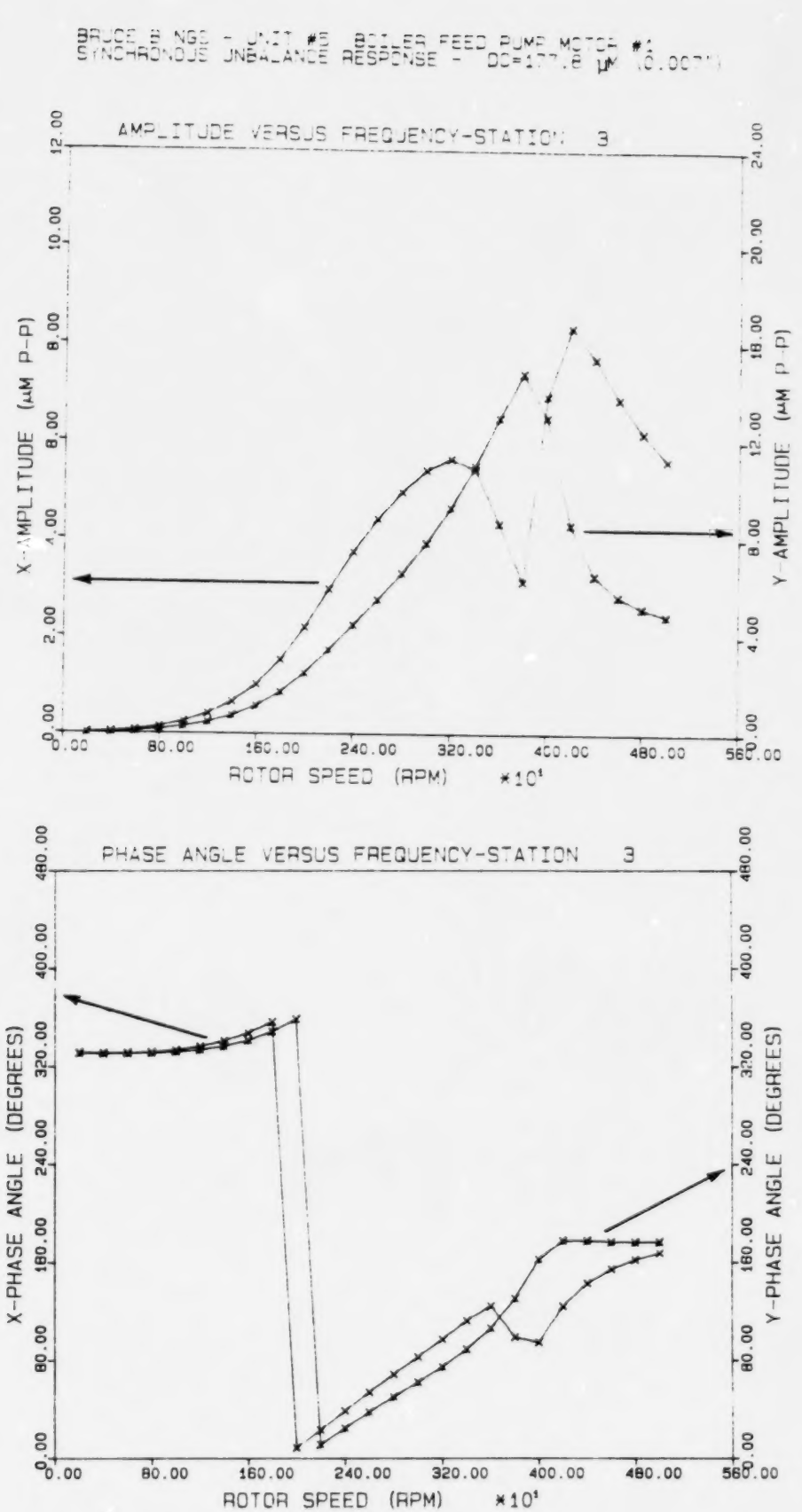

BRUCE B NGS - UNIT #8 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=177.8 µM (0.007")
AMPLITUDE VERSUS FREQUENCY-STATION 3
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 3
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10¹

Figure 18

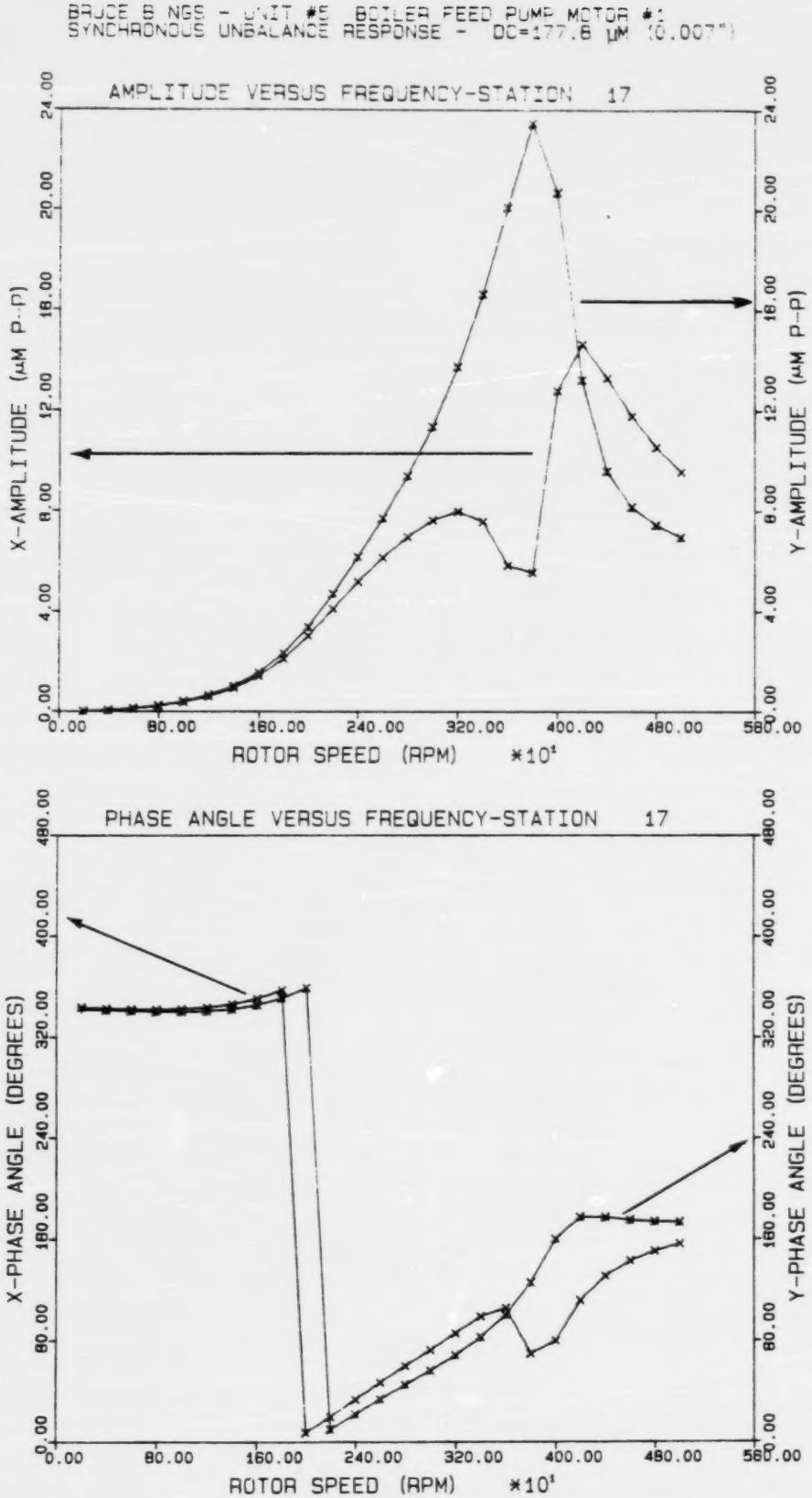

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=177.8 µM (0.007")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
160.00
240.00
320.00
400.00
480.00
560.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 19

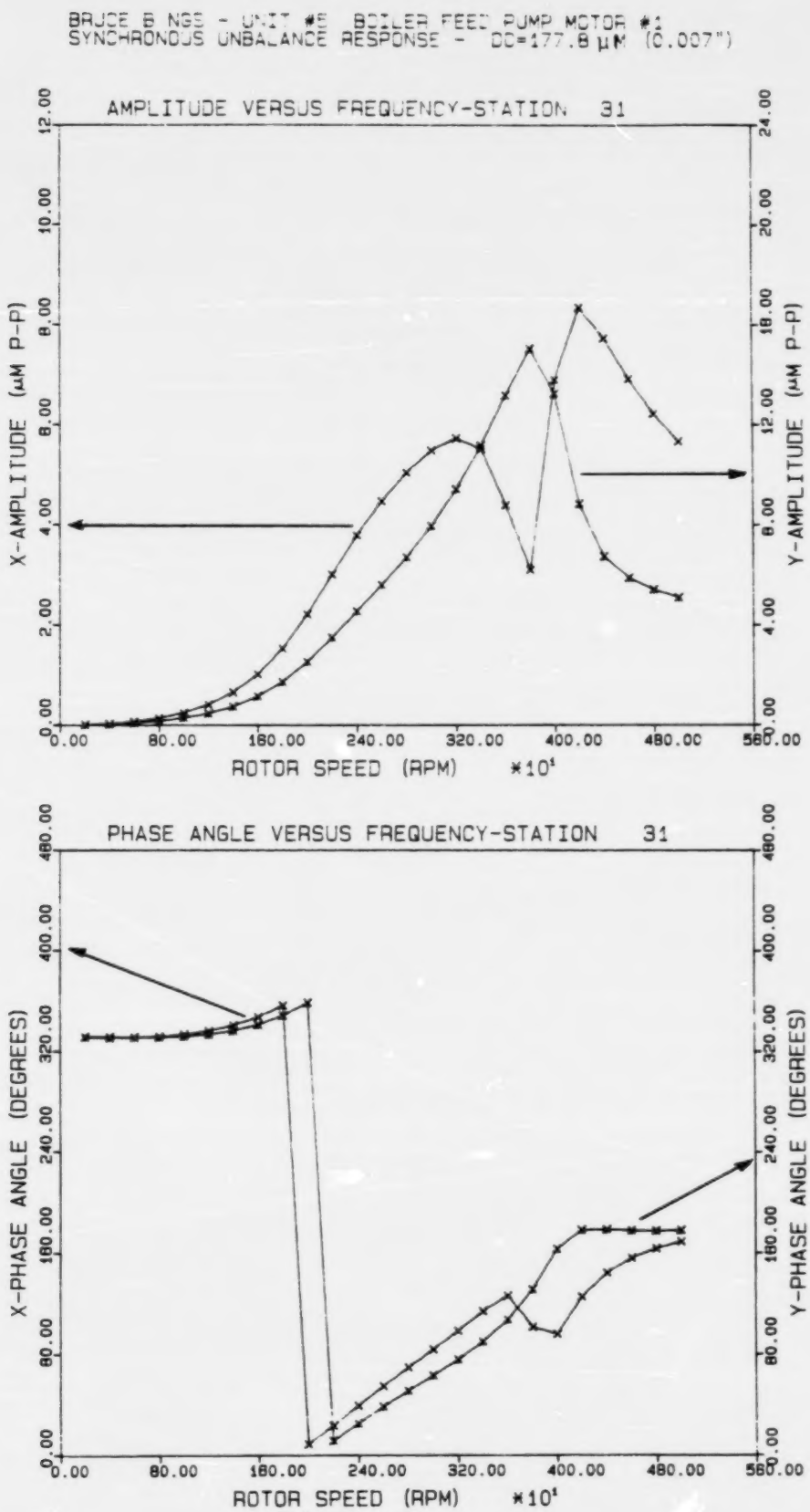

BRUCE B NGS - UNIT #8 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=177.8 µM (0.007")
AMPLITUDE VERSUS FREQUENCY-STATION 31
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 31
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10¹

Figure 20

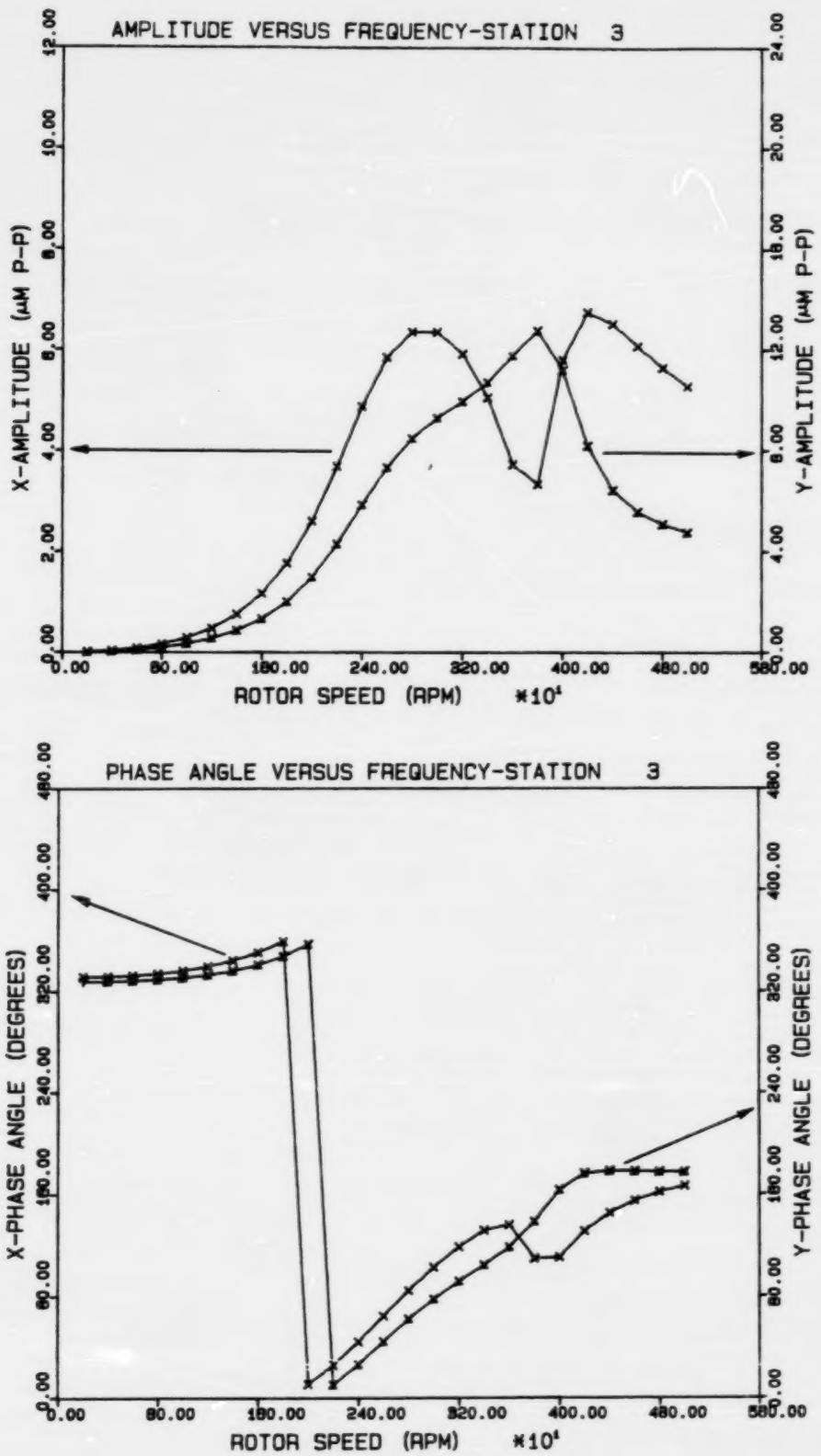

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=203.2 μM (0.008")
AMPLITUDE VERSUS FREQUENCY-STATION 3
X-AMPLITUDE (μM P-P)
Y-AMPLITUDE (μM P-P)
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 3
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10¹

Figure 21

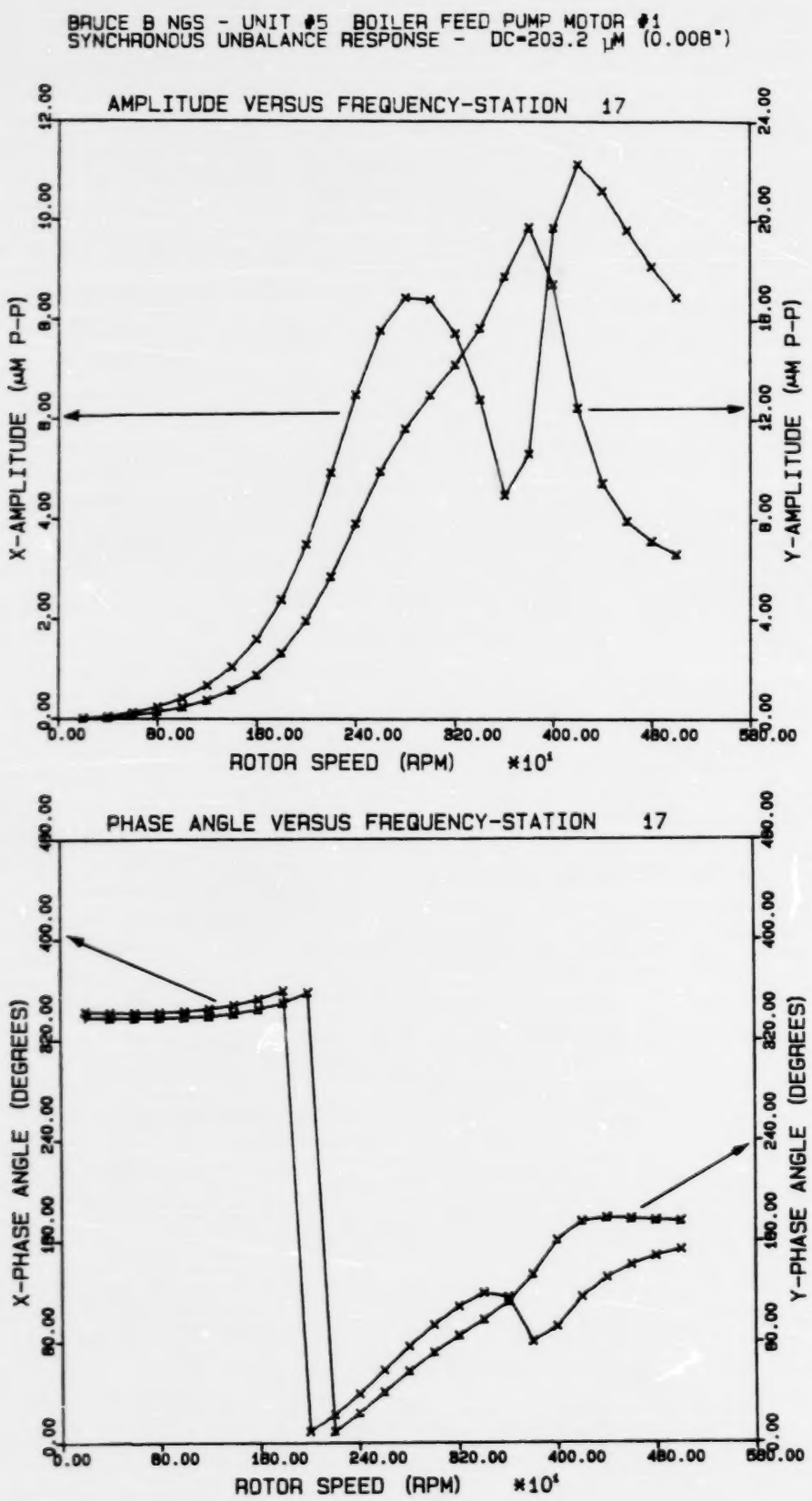
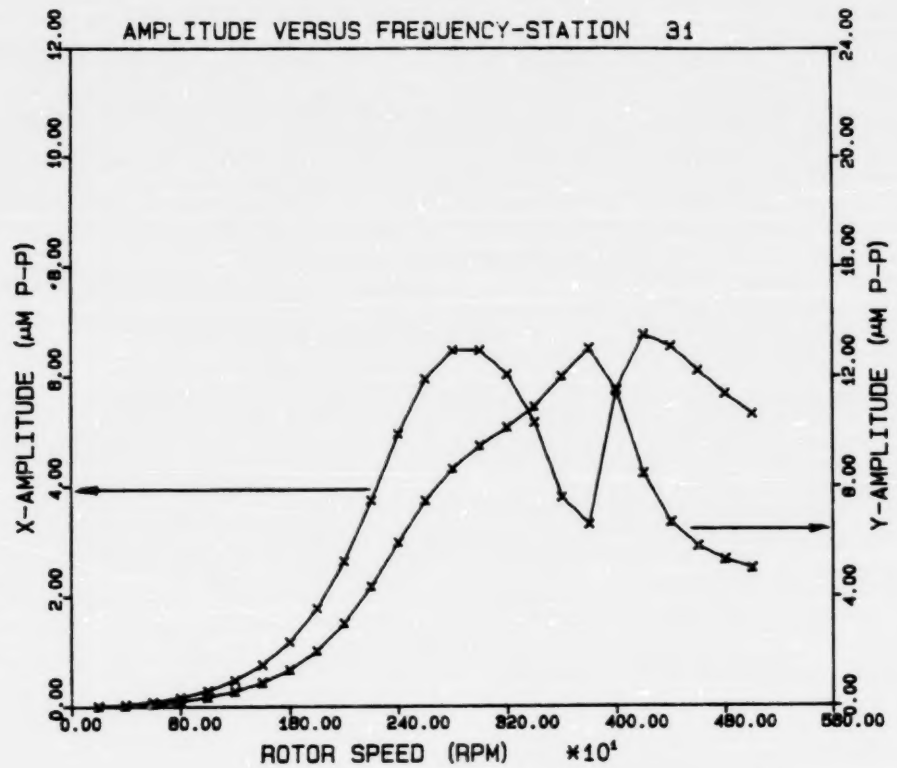

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=203.2 μM (0.008")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (μM P-P)
Y-AMPLITUDE (μM P-P)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
160.00
240.00
320.00
400.00
480.00
560.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
0.00
80.00
160.00
240.00
320.00
400.00
480.00
0.00
80.00
160.00
240.00
320.00
400.00
480.00
560.00
ROTOR SPEED (RPM) *10¹

Figure 22

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=203.2 µM (0.008")



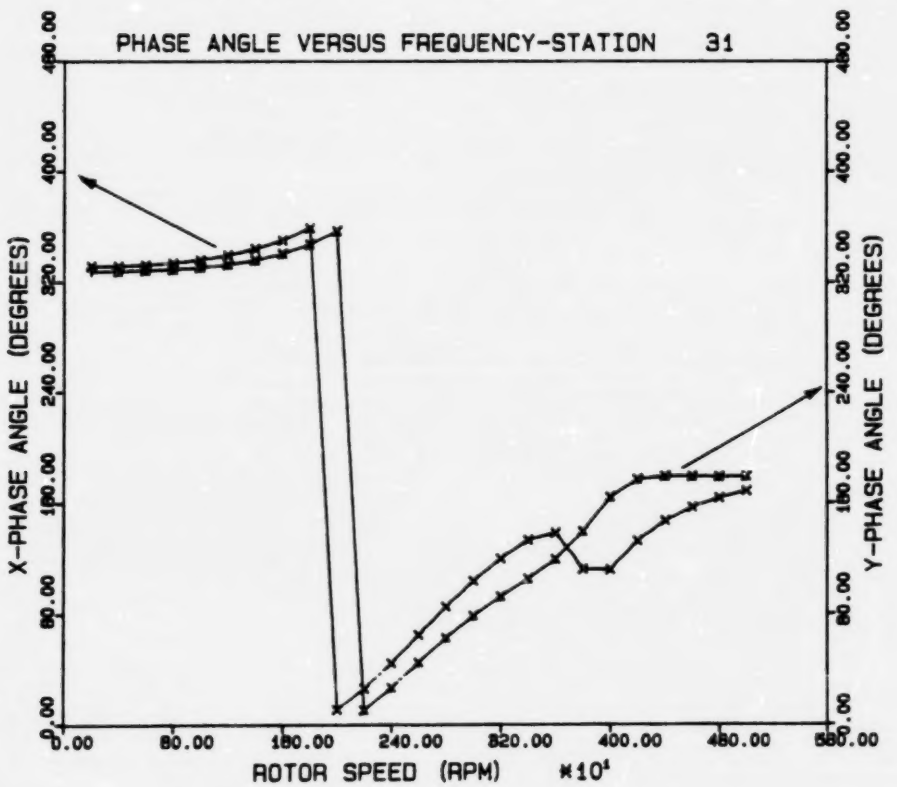
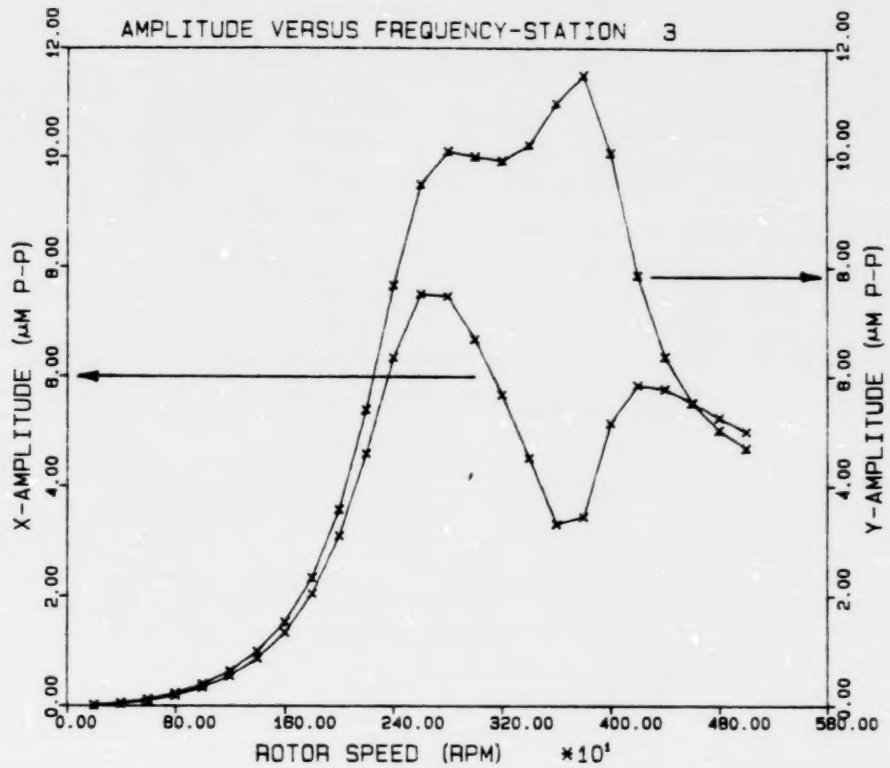


Figure 23

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=228.6 µM (0.009")



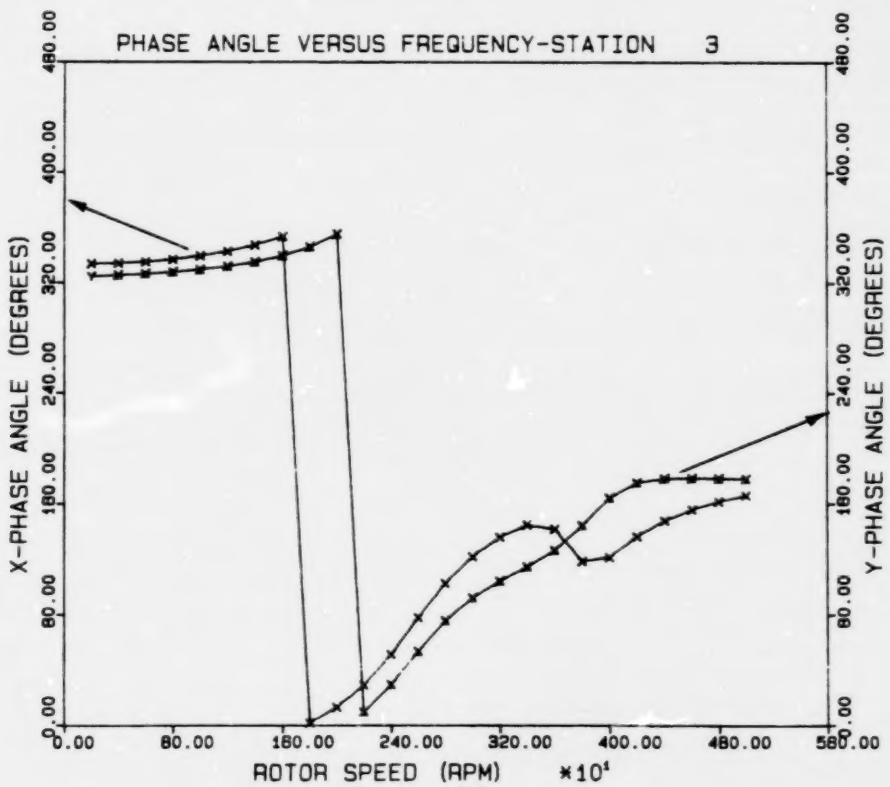


Figure 24

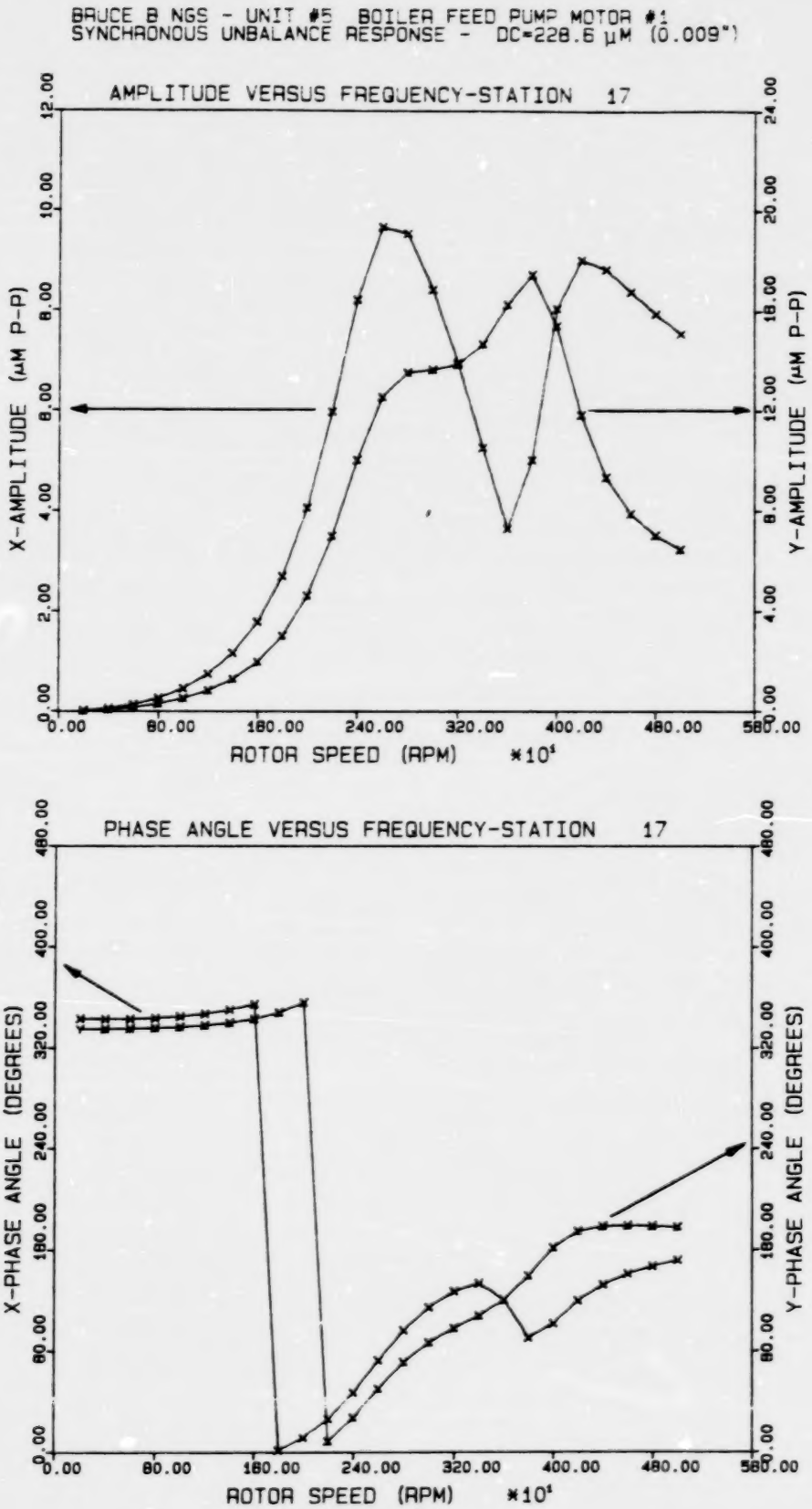

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=228.6 µM (0.009")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
160.00
240.00
320.00
400.00
480.00
560.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 25

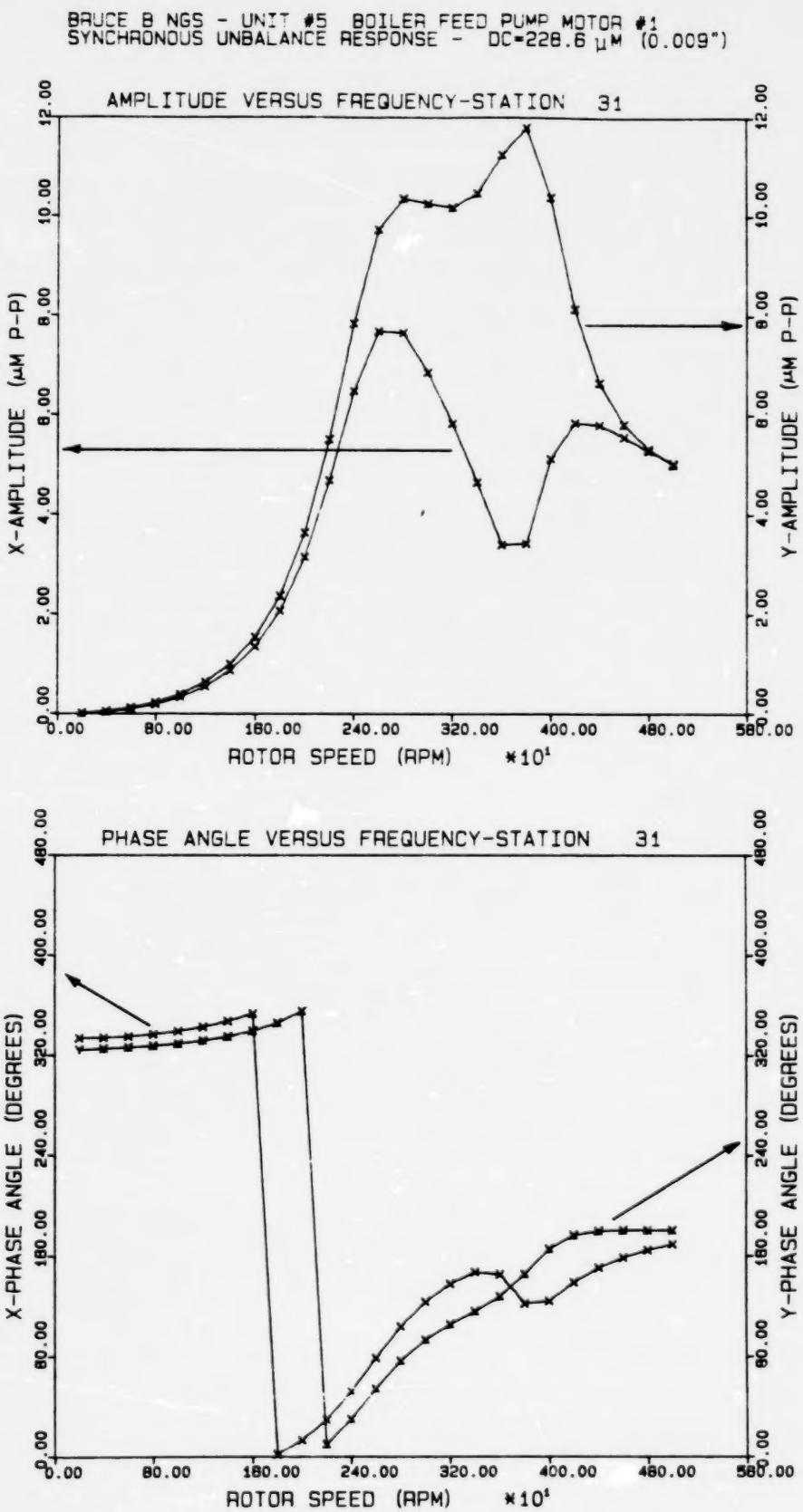
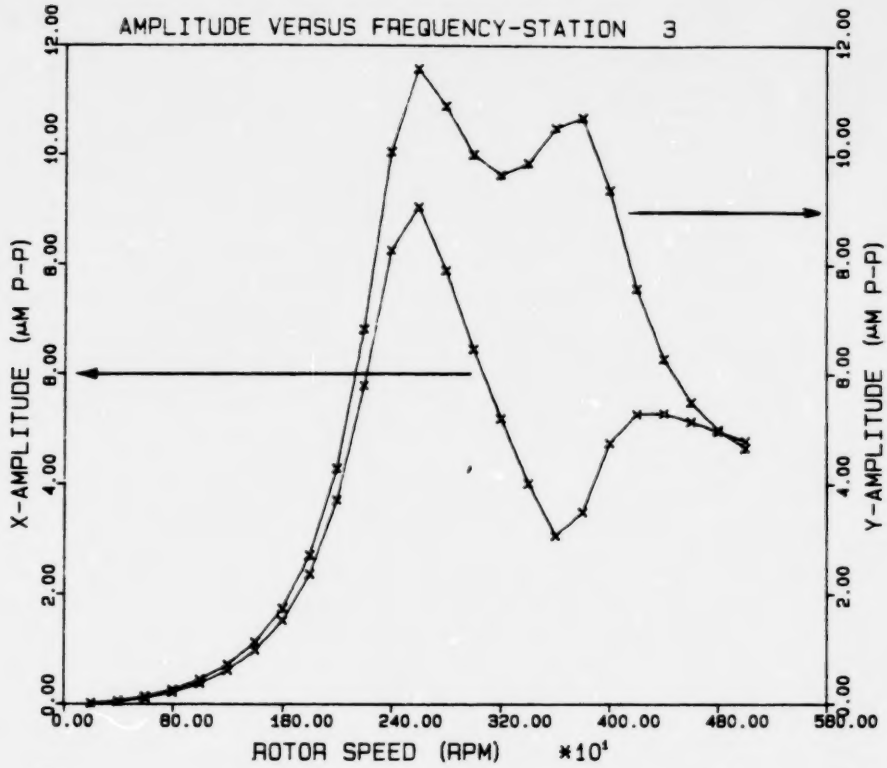

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=228.6 µM (0.009")
AMPLITUDE VERSUS FREQUENCY-STATION 31
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 31
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
180.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 26

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=254.0 µM (0.010")



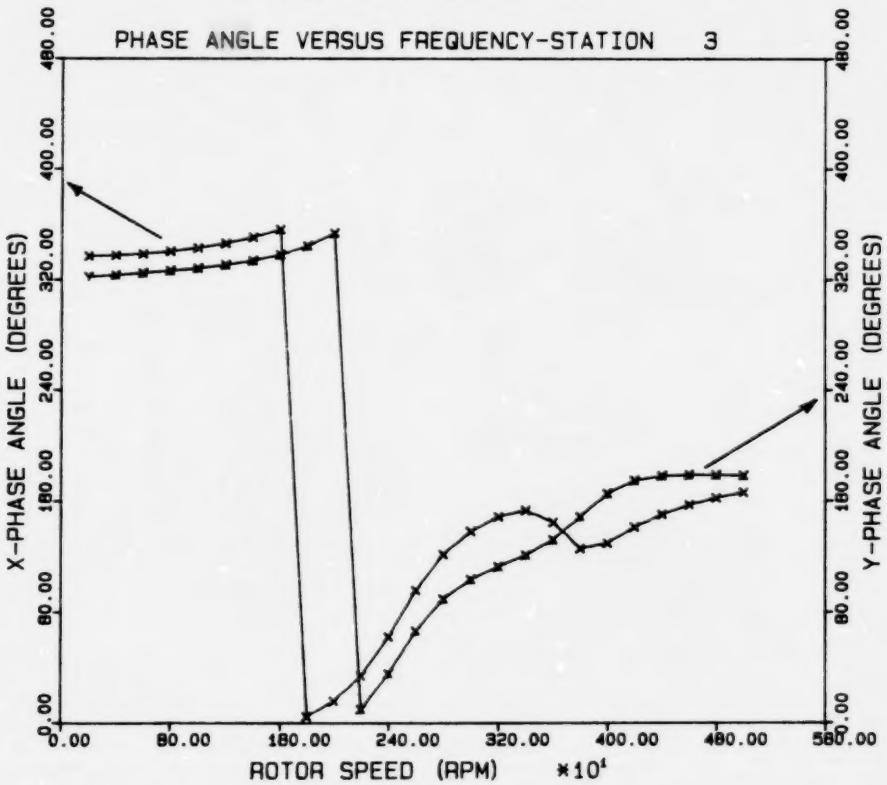


Figure 27

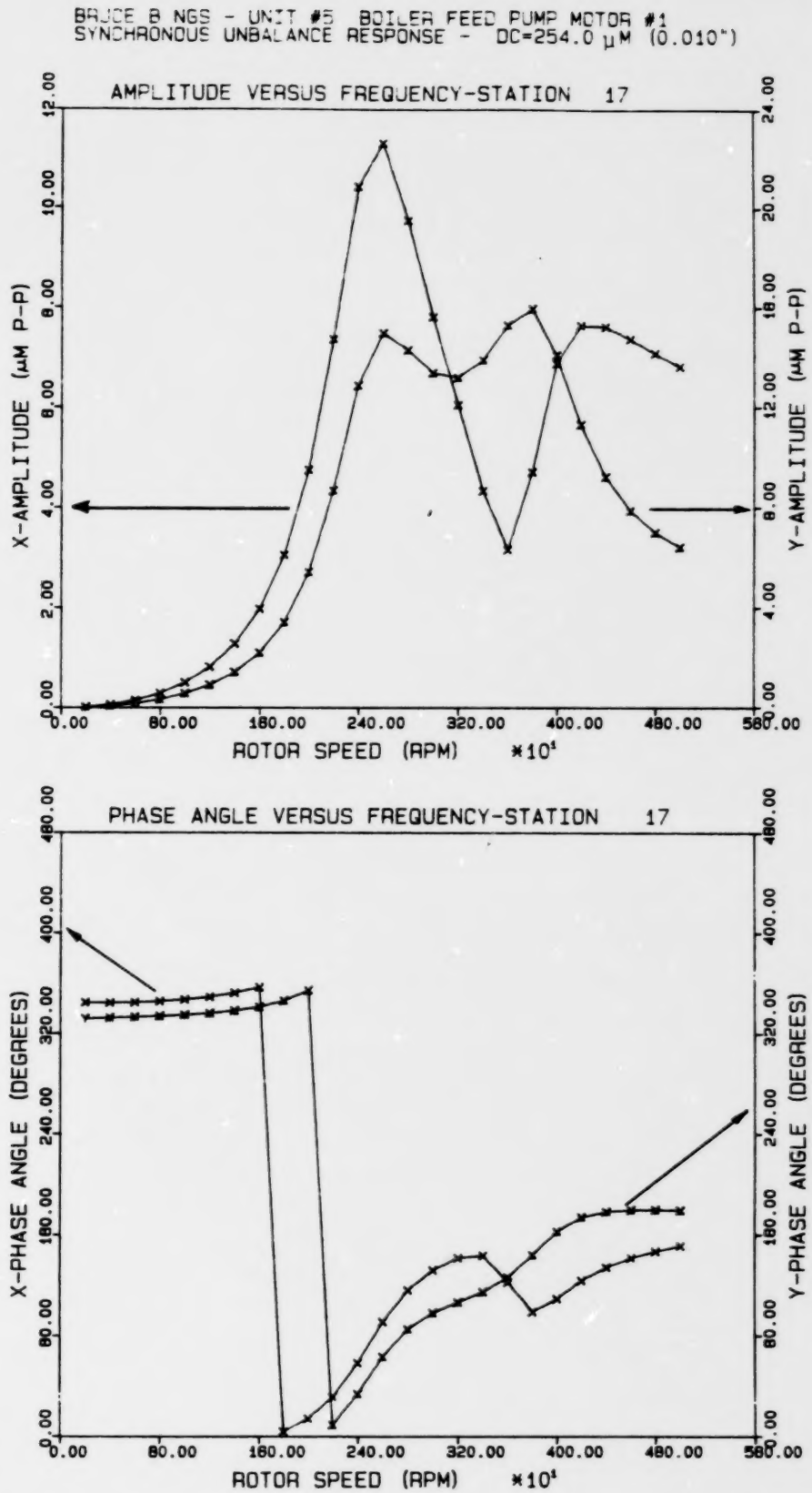
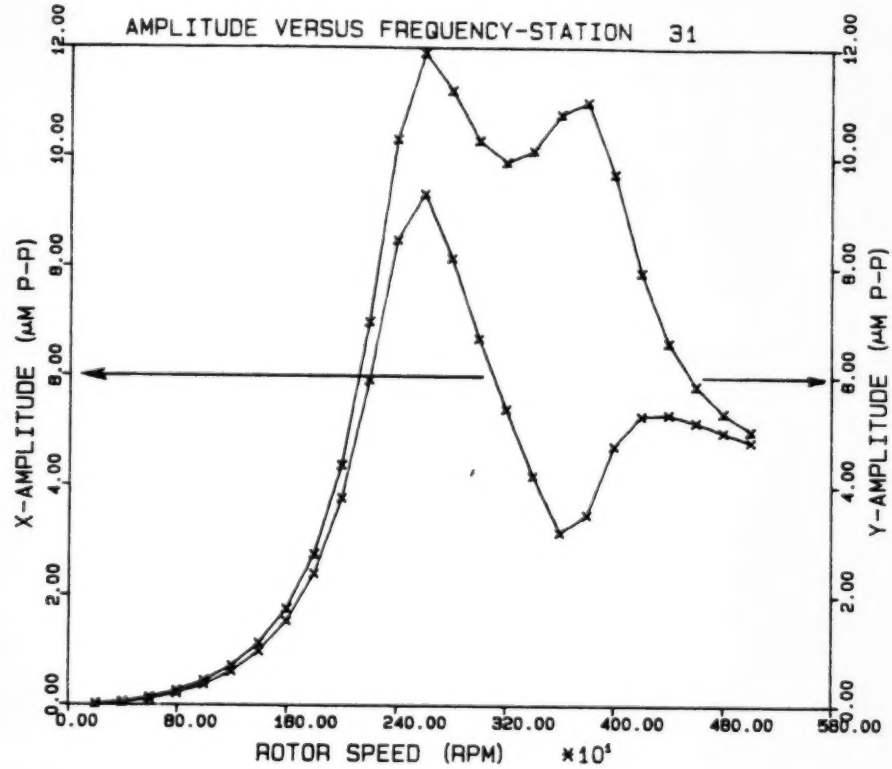

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=254.0 µM (0.010")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
2.00
4.00
6.00
8.00
10.00
12.00
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
160.00
240.00
320.00
400.00
480.00
560.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 28

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=254.0 µM (0.010")



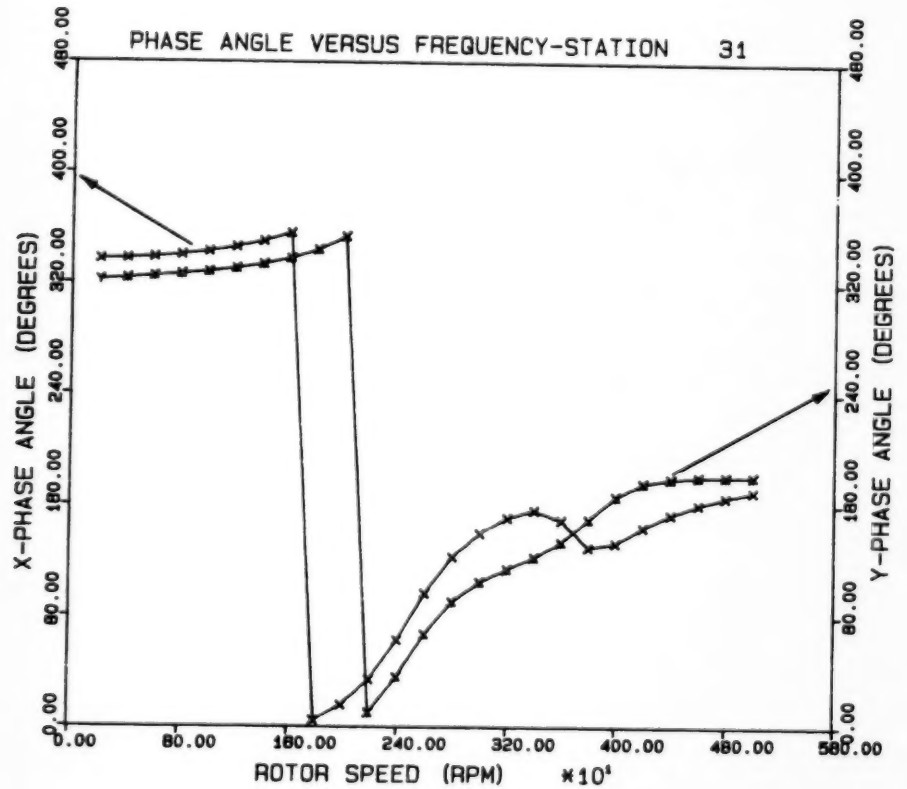


Figure 29

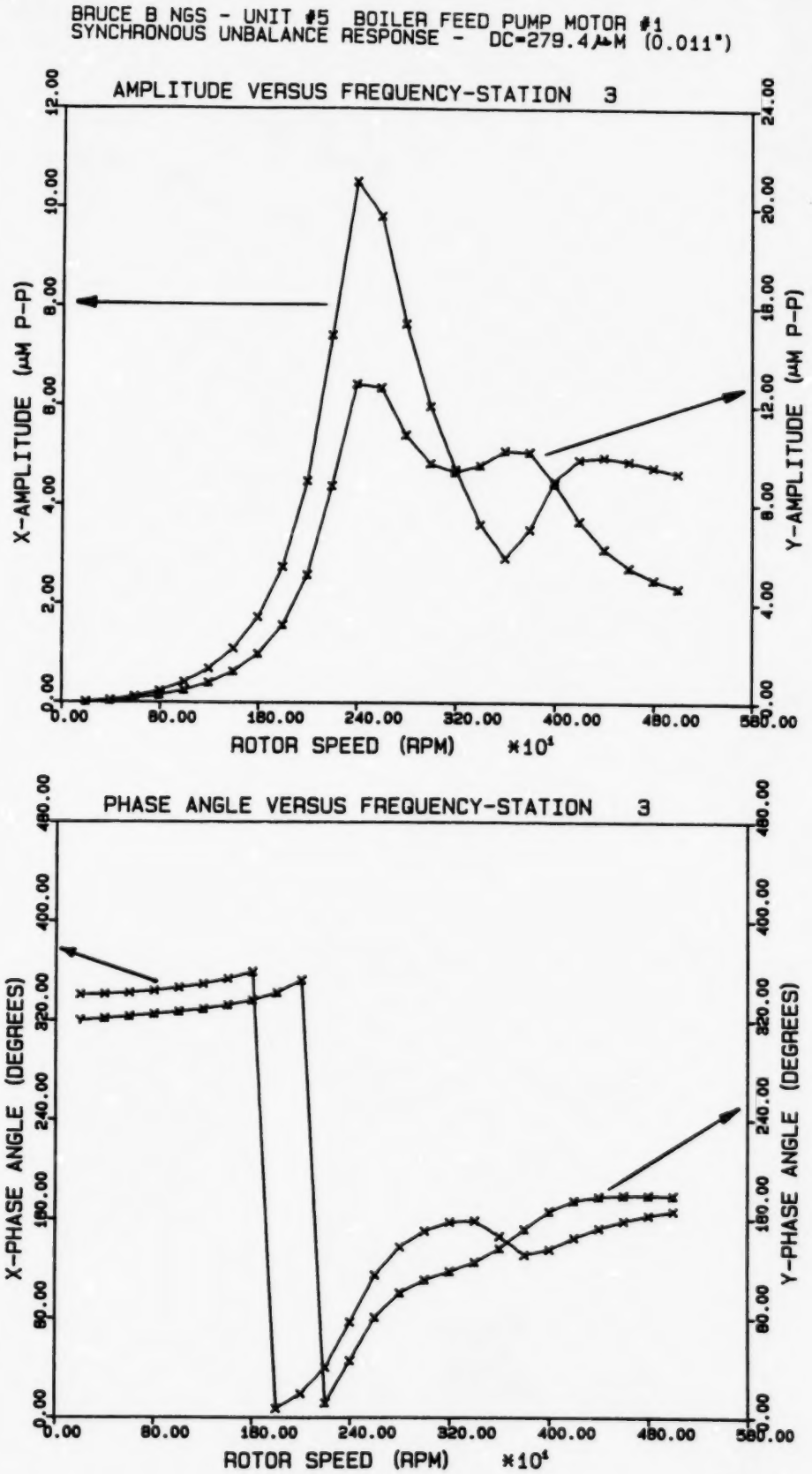
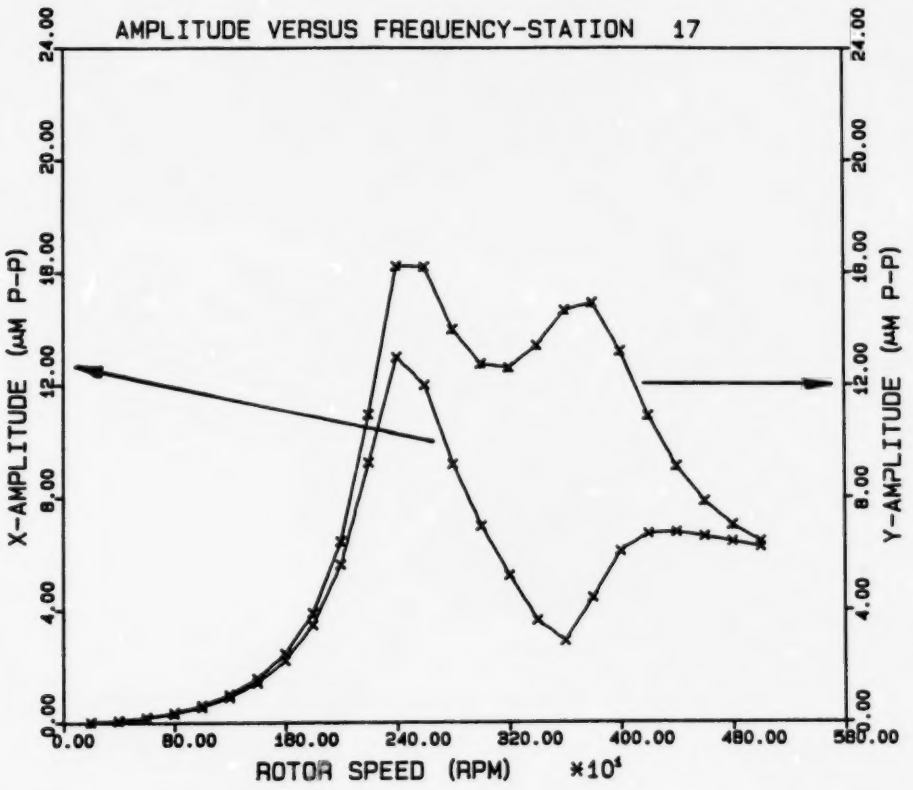

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=279.4μM (0.011")
AMPLITUDE VERSUS FREQUENCY-STATION 3
X-AMPLITUDE (μM P-P)
Y-AMPLITUDE (μM P-P)
ROTOR SPEED (RPM) *10⁴
PHASE ANGLE VERSUS FREQUENCY-STATION 3
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10⁴

Figure 30

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=279.4μM (0.011")



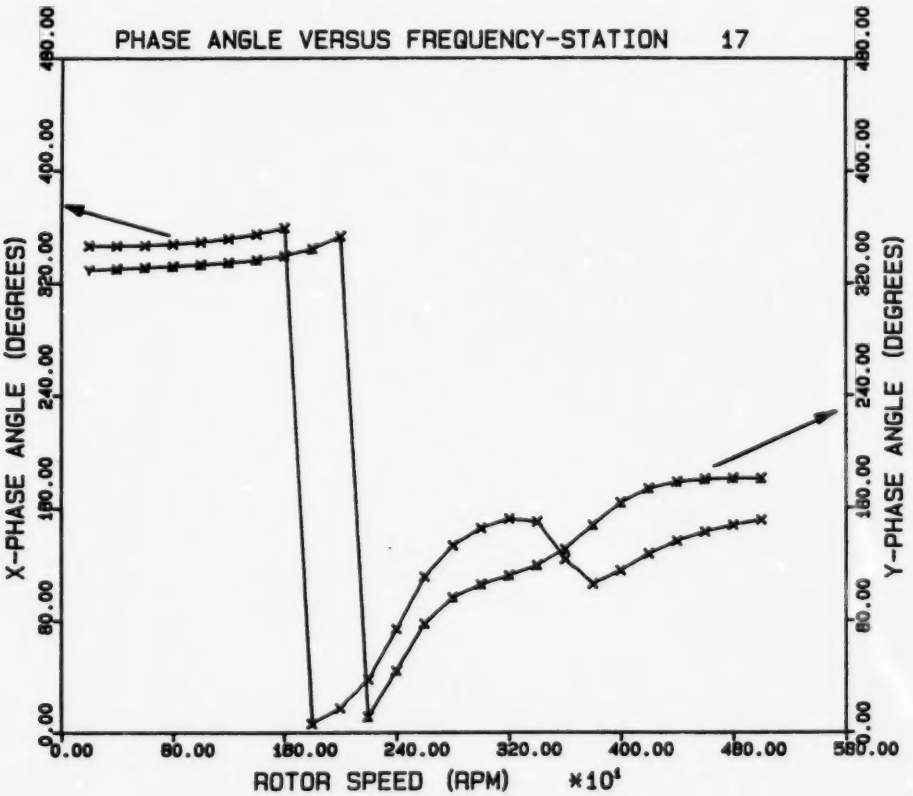


Figure 31

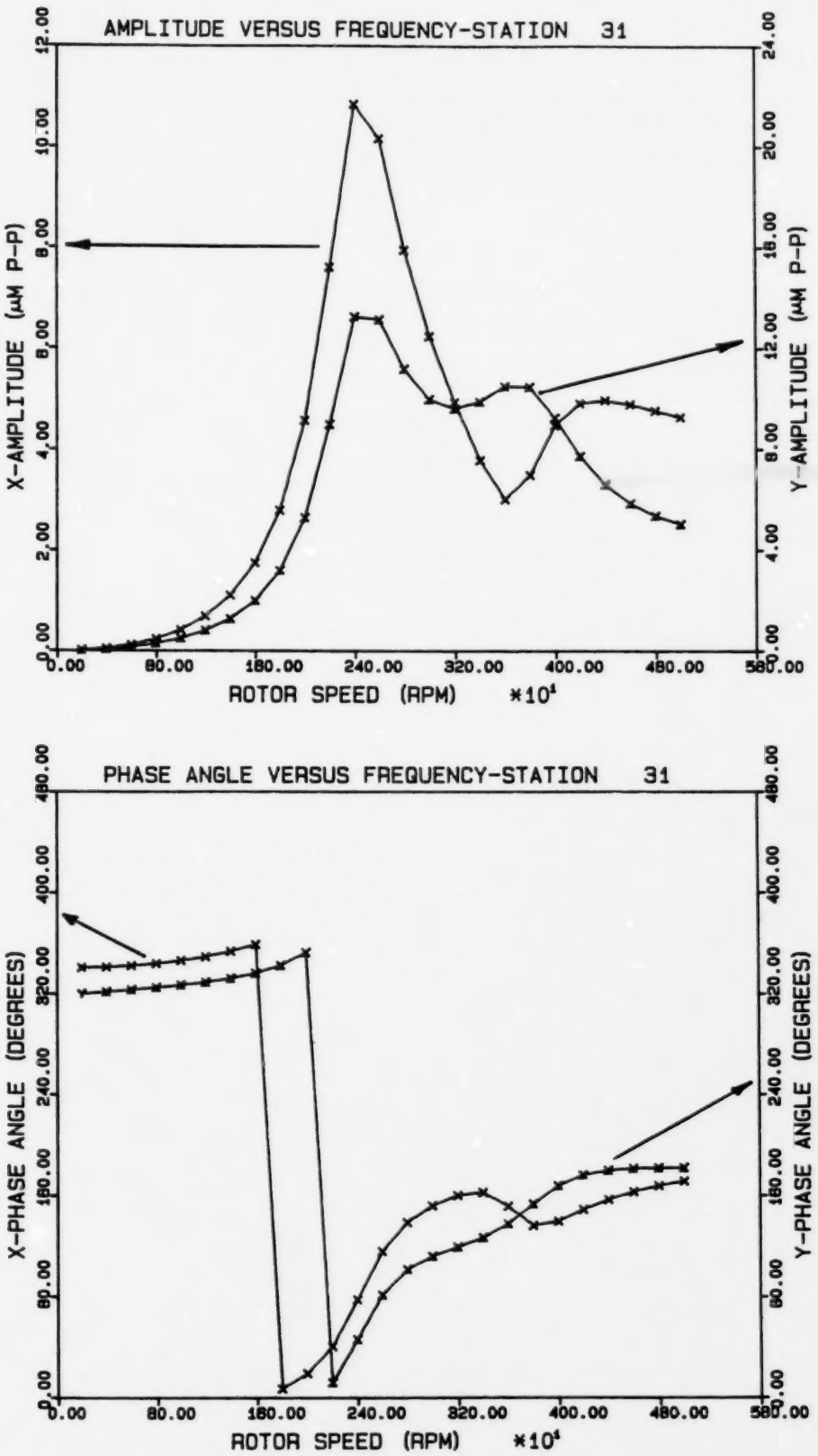

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=279.4μM (0.011")
AMPLITUDE VERSUS FREQUENCY-STATION 31
X-AMPLITUDE (μM P-P)
Y-AMPLITUDE (μM P-P)
ROTOR SPEED (RPM) *10⁴
PHASE ANGLE VERSUS FREQUENCY-STATION 31
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
ROTOR SPEED (RPM) *10⁴

Figure 32

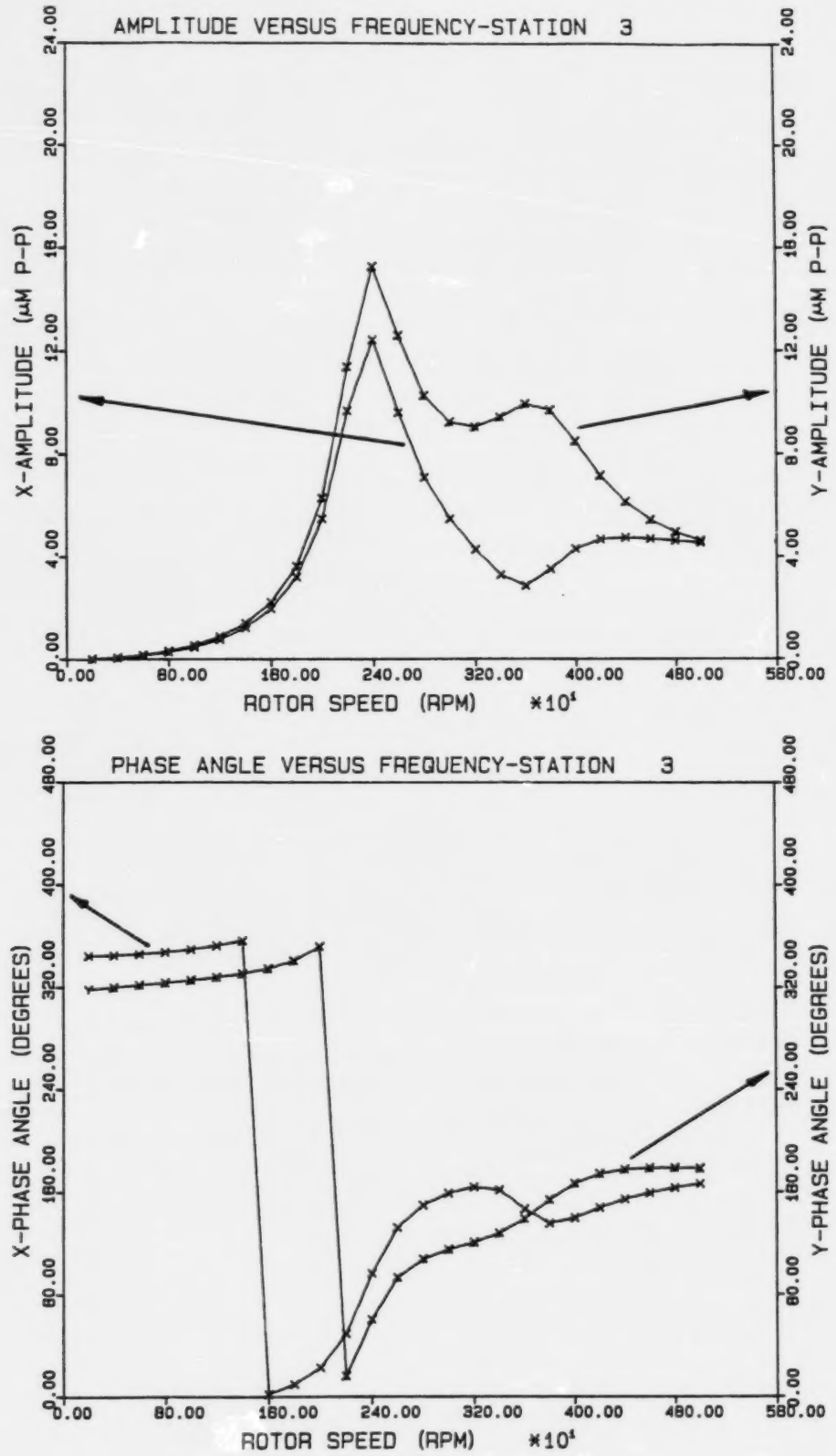
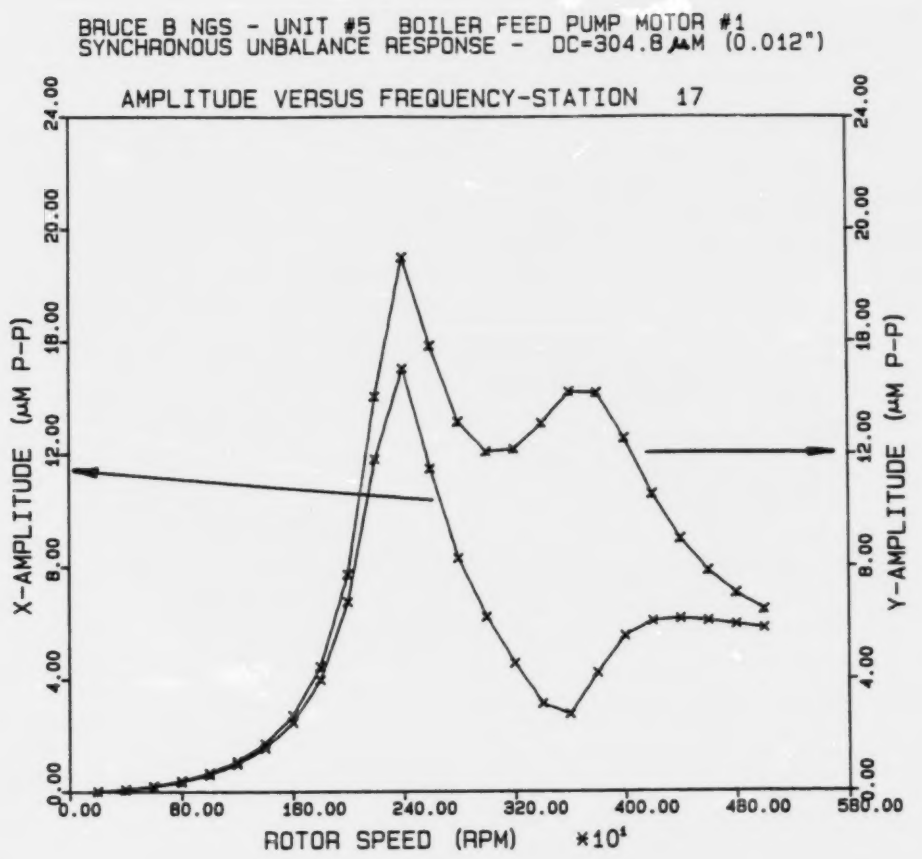

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=304.8 µM (0.012")
AMPLITUDE VERSUS FREQUENCY-STATION 3
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 3
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 33


BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=304.8 µM (0.012")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹

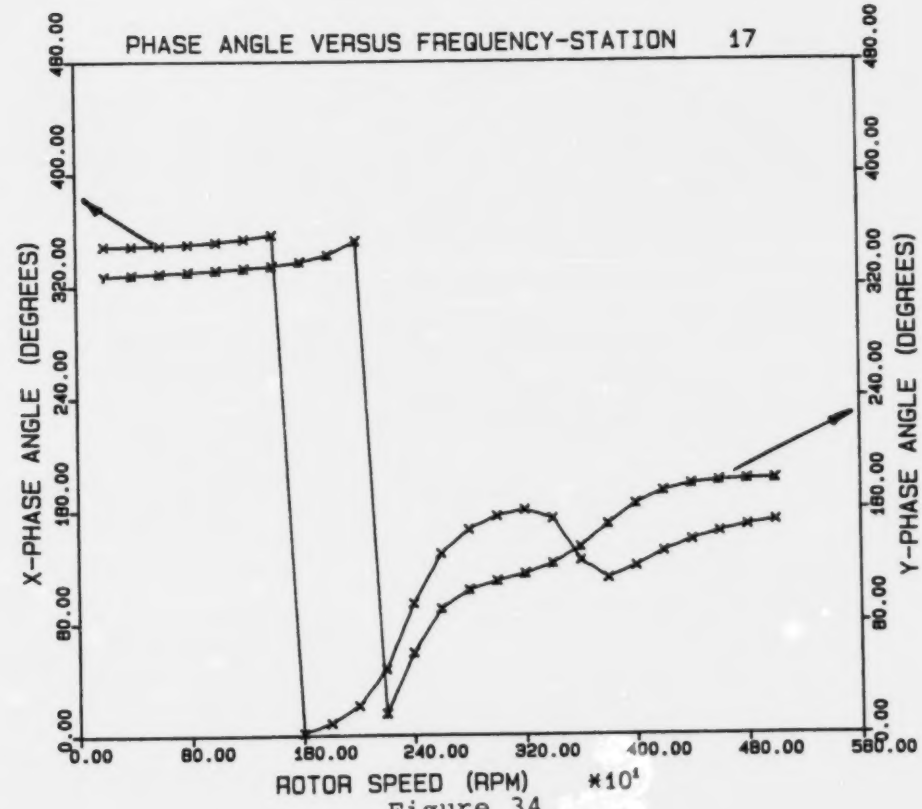
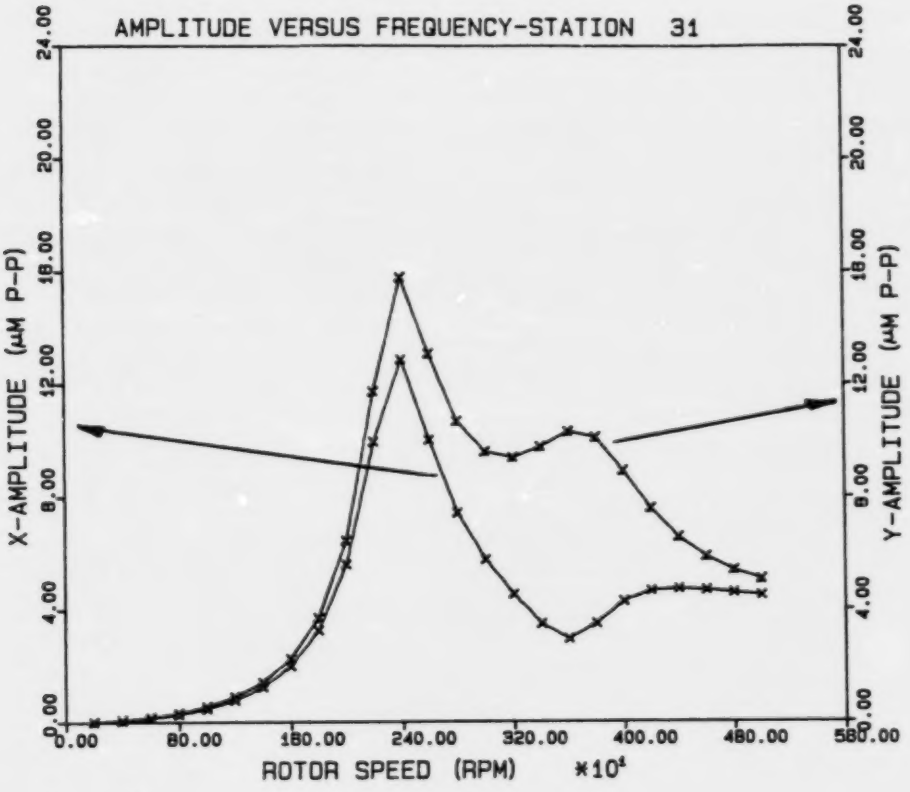

PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹

Figure 34

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=304.8 μM (0.012")



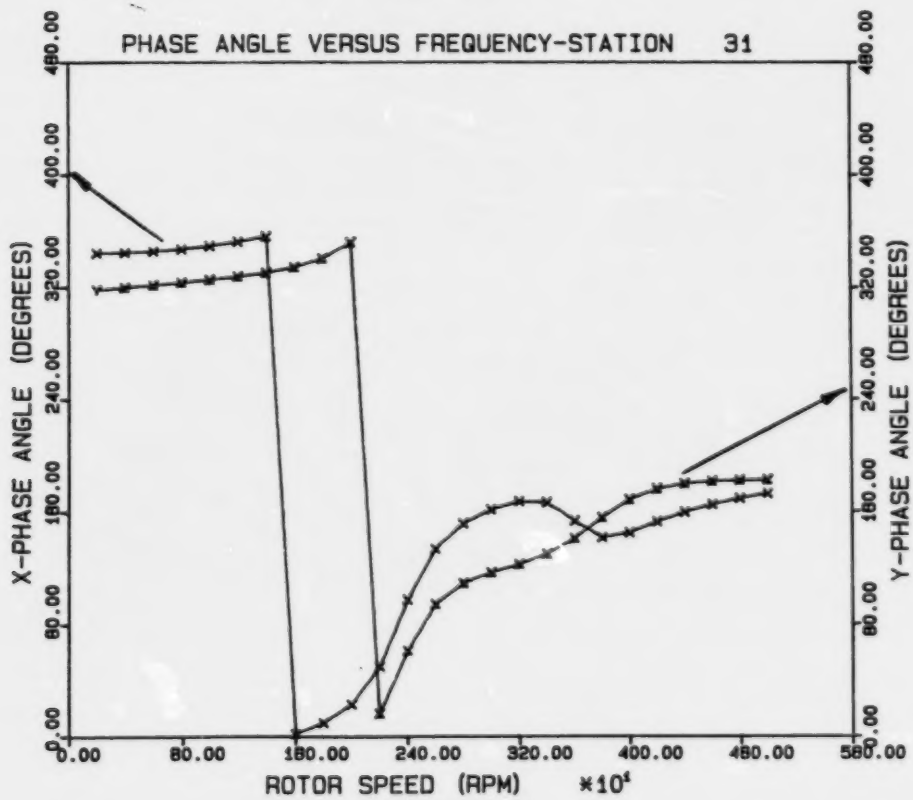


Figure 35

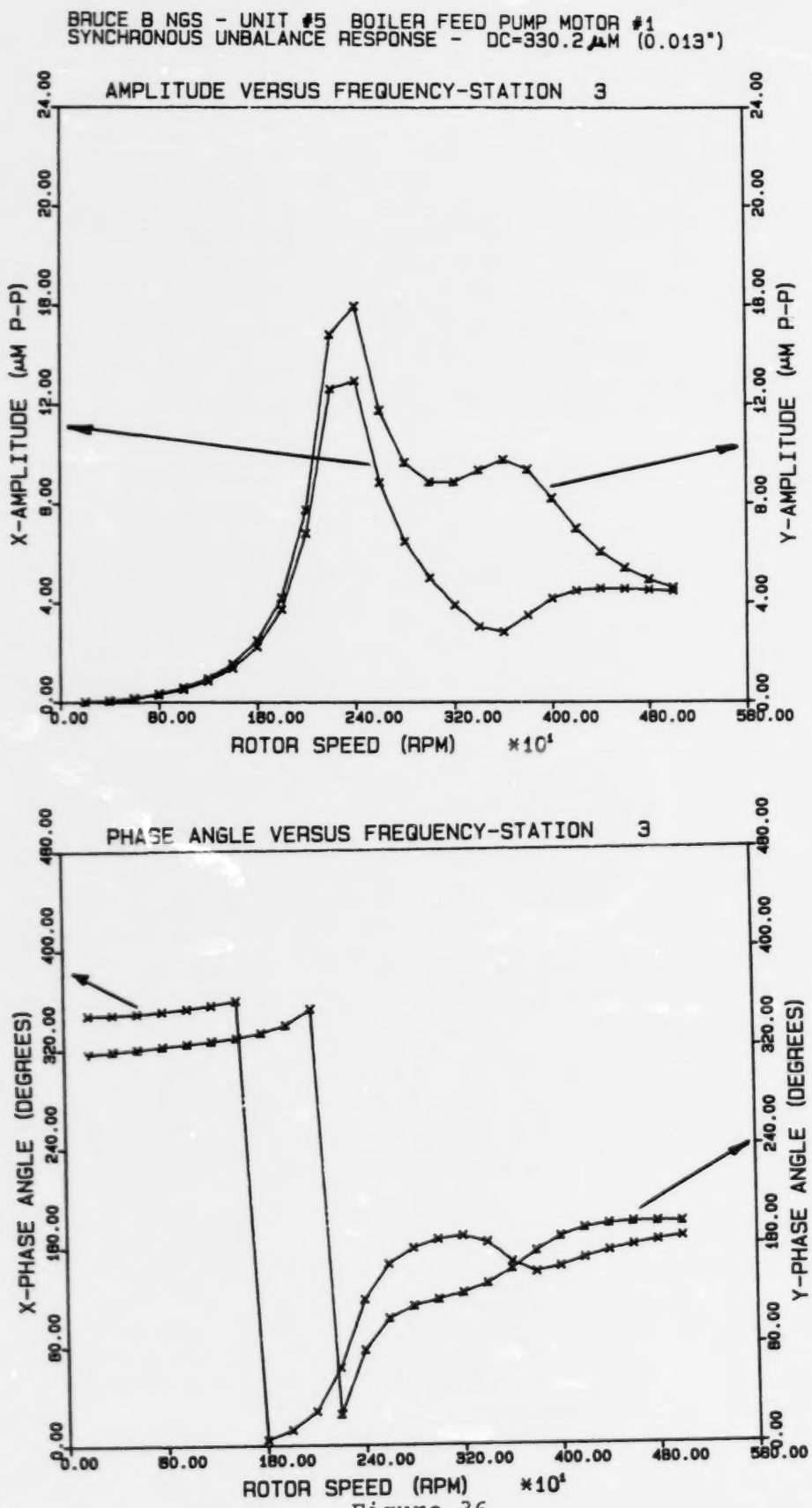

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=330.2 µM (0.013")
AMPLITUDE VERSUS FREQUENCY-STATION 3
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 3
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹

Figure 36

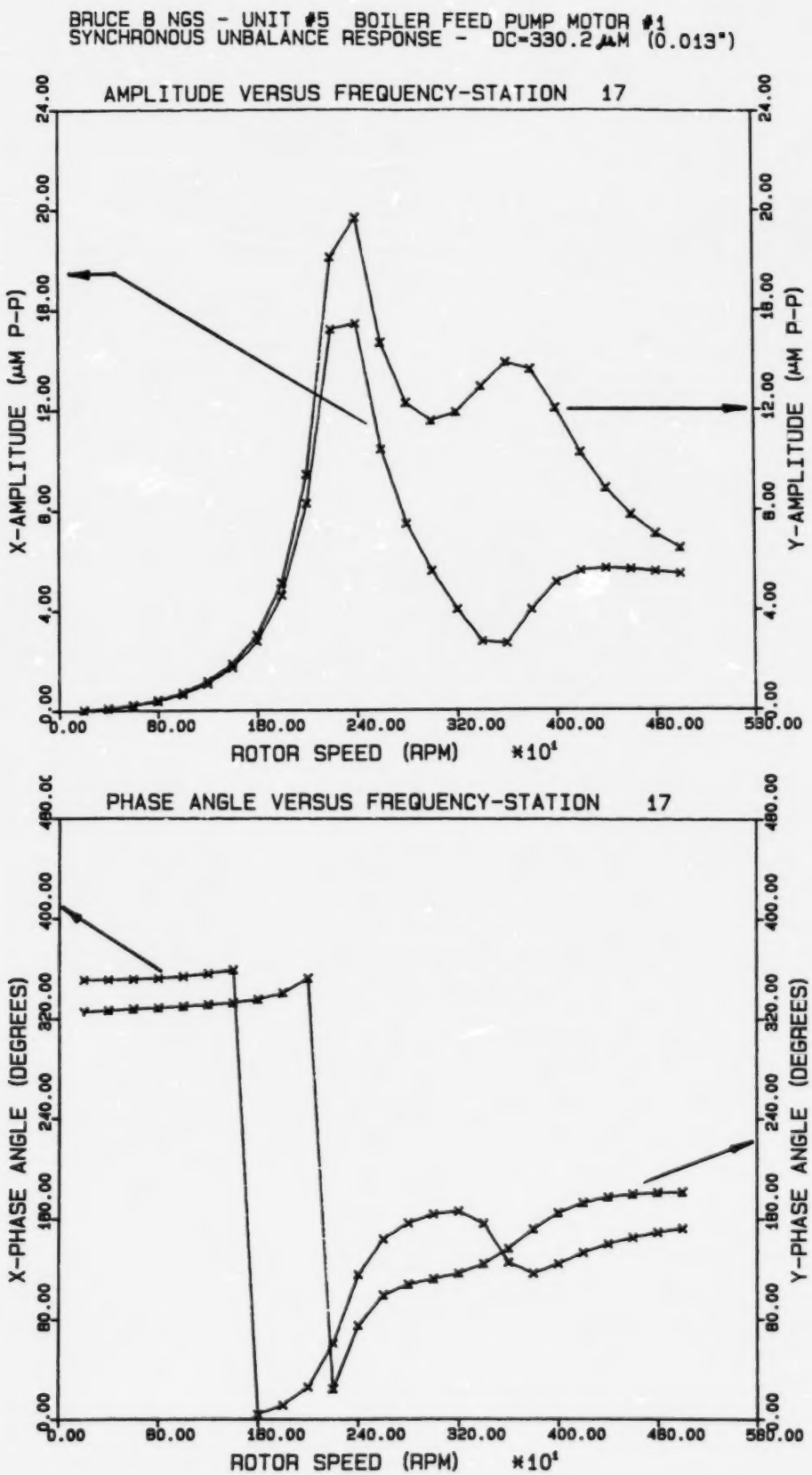

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=330.2 µM (0.013")
AMPLITUDE VERSUS FREQUENCY-STATION 17
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
560.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 17
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 37

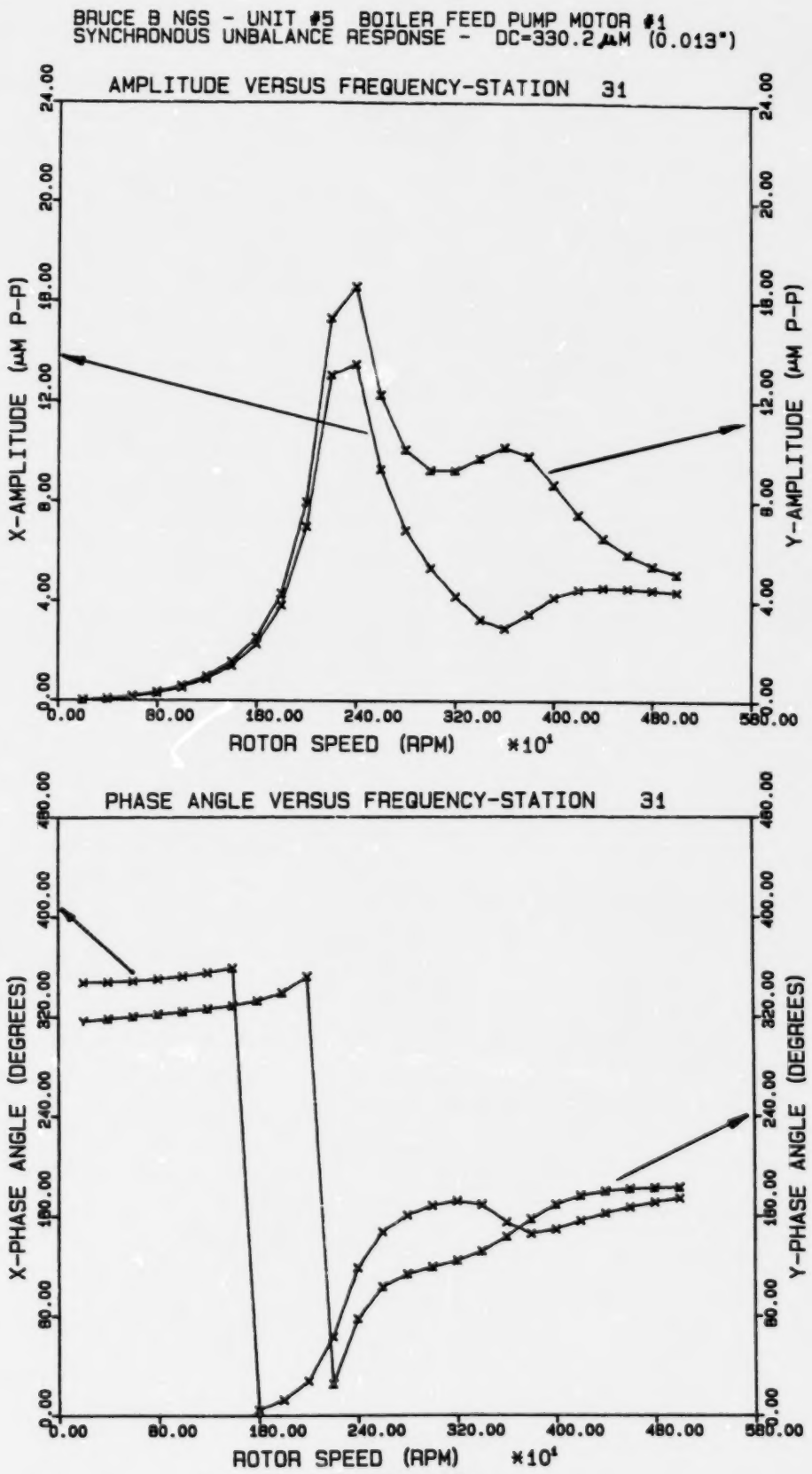

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=330.2 µM (0.013")
AMPLITUDE VERSUS FREQUENCY-STATION 31
X-AMPLITUDE (µM P-P)
Y-AMPLITUDE (µM P-P)
0.00
4.00
8.00
12.00
16.00
20.00
24.00
0.00
80.00
180.00
240.00
320.00
400.00
480.00
580.00
ROTOR SPEED (RPM) *10¹
PHASE ANGLE VERSUS FREQUENCY-STATION 31
X-PHASE ANGLE (DEGREES)
Y-PHASE ANGLE (DEGREES)
0.00
80.00
160.00
240.00
320.00
400.00
480.00
ROTOR SPEED (RPM) *10¹

Figure 38

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=203.2 µM (0.008")

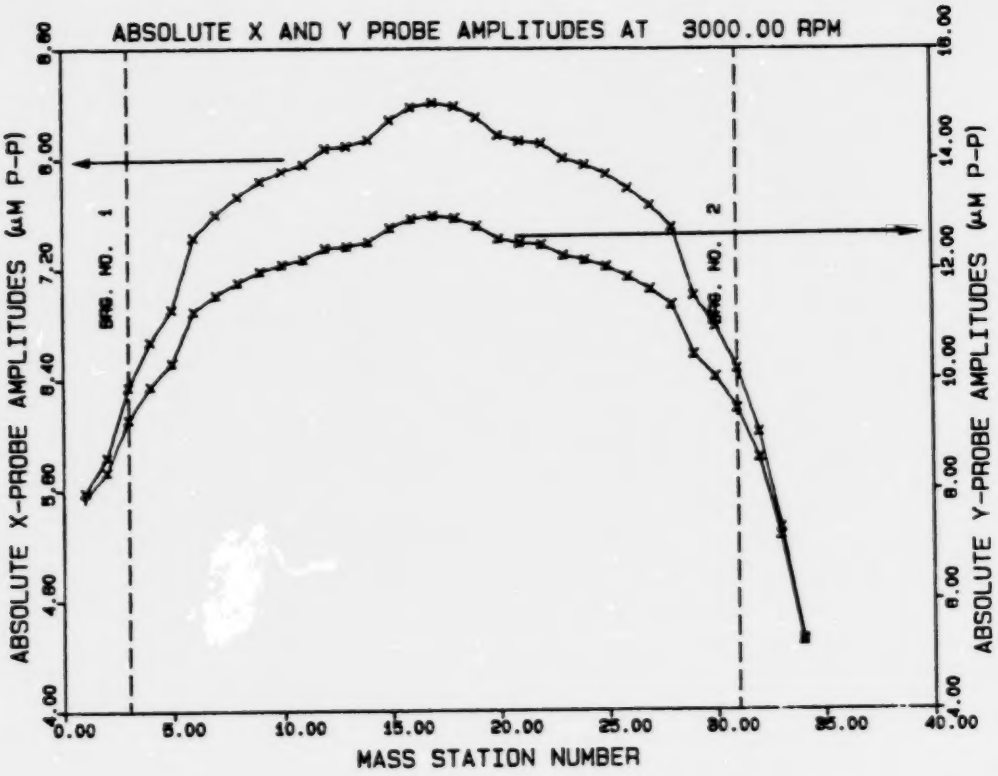


Figure 39

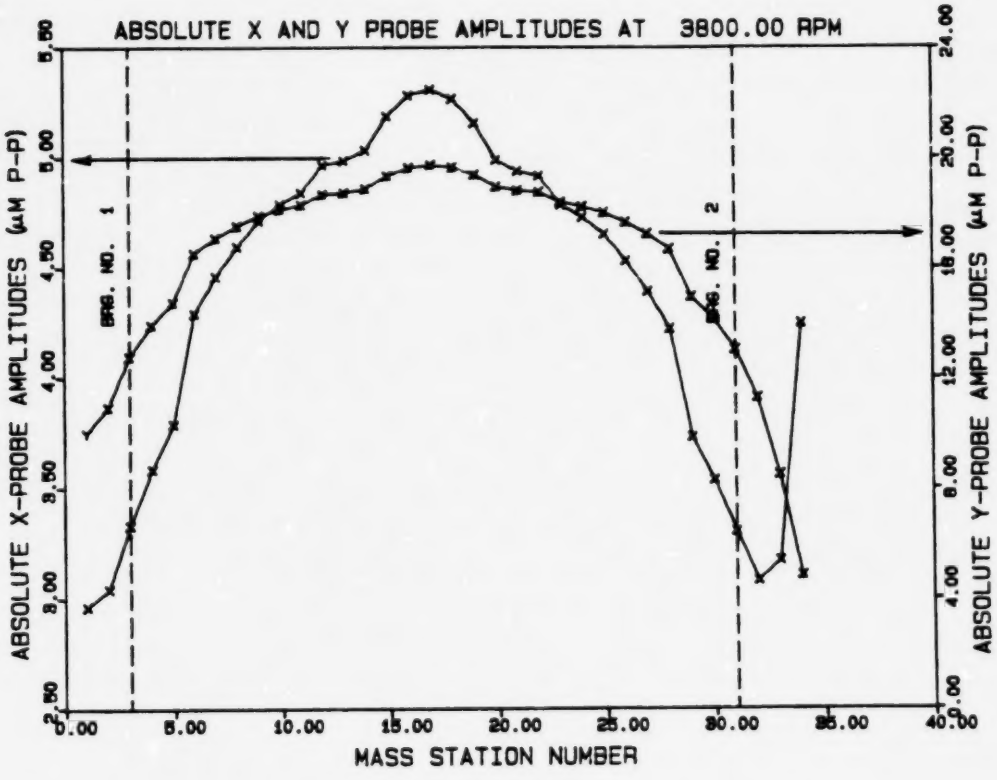
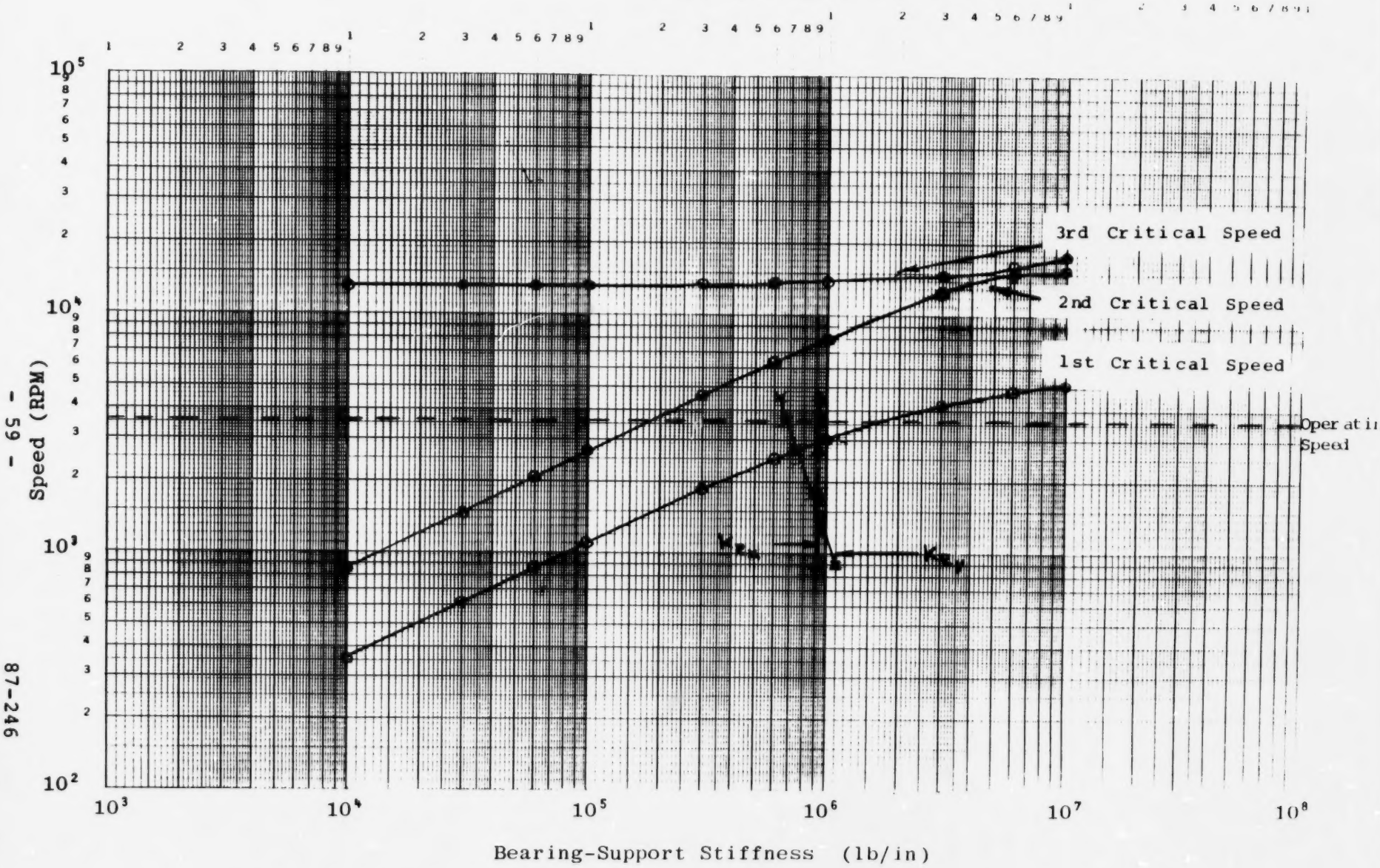

BRUCE B NGS - UNIT #5 BOILER FEED PUMP MOTOR #1
SYNCHRONOUS UNBALANCE RESPONSE - DC=203.2 µM (0.008")
ABSOLUTE X AND Y PROBE AMPLITUDES AT 3800.00 RPM
ABSOLUTE X-PROBE AMPLITUDES (µM P-P)
ABSOLUTE Y-PROBE AMPLITUDES (µM P-P)
BRG. NO. 1
BRG. NO. 2
MASS STATION NUMBER
0.00
5.00
10.00
15.00
20.00
25.00
30.00
35.00
40.00

Figure 40

K·E LOGARITHMIC 3 x 5 CYCLES
KEUFFEL & ESSER CO MADE IN U.S.A.
46 7520

FIGURE 41 : Bruce NGS 'B' Unit #5- Boiler Feed Pump Motor #1
Critical Speed Map



87-246

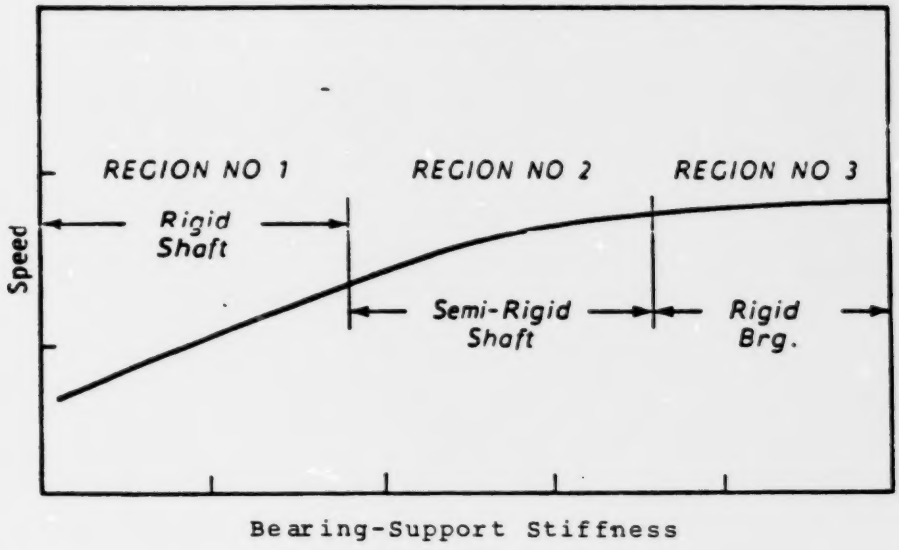


FIGURE 42

TYPICAL CRITICAL SPEED MAP FOR THE FIRST MODE

APPENDIX A

RELATIVE AND ABSOLUTE DISPLACEMENT MEASUREMENTS

Experimental rotor dynamics analyses are sometimes complicated by the fact that the only available location for the installation of displacement transducers is the bearing housing which can be located near rotor nodes (especially for the first mode). Thus rotor vibration can be very small and can be easily missed. In addition, if the casing vibrates in phase with the rotor and the amplitudes are of the same magnitude, the rotor displacement signal will be close to zero since the displacement transducer measures the relative displacement between the rotor and casing.

A method of enhancing the rotor vibration is to measure the absolute rotor vibration. This is accomplished by installing a velocity or an accelerometer transducer on the casing in the same direction as the displacement transducer performing the appropriate number of integrations to obtain displacement and carrying out a vector addition of the casing and rotor displacement vibrations.

APPENDIX B

ALLOWABLE VIBRATION LIMITS FOR ROTATING MACHINERY AT BRUCE NGS 'B' NUCLEAR POWER STATION

CHART 5.1

ALLOWABLE VIBRATION LIMITS FOR ROTATING MACHINERY

For commissioning purposes the standard measurement location(s) shall be all bearings (driver and driven equipment) for horizontal machines, and the top motor bearing for vertical machines. Where casing vibration is of concern, the same limits shall apply to the maximum vibration measured at any location.

Class of Machines*	Bearing Radial or Axial Vibration Velocity Level, Unfiltered, mm/s, Zero-to-Peak								
	Horizontal or Vertical Machines (Up to 1.5 m High)+			Vertical Machines (Between 1.5 and 3 m High)+			Vertical Machines (Between 3 and 6 m High)+		
	Smooth	Limit of Acceptance	Do Not Exceed	Smooth	Limit of Acceptance	Do Not Exceed	Smooth	Limit of Acceptance	Do Not Exceed
Class A	1.0	3.5	6.4	2.0	3.5	7.4	3.0	4.5	8.9
Class B	1.6	4.0	10.0	2.5	5.0	11.0	3.5	5.5	12.5
Class C&D	2.0	5.0	15.0	3.0	6.0	17.0	4.0#	6.5#	18.0#

*Class A: Small machines up to 15 kW, e.g. oil pumps in PHT main circuit (3312)

Class B: Medium sized machines up to 300 kW, e.g. maintenance cooling pumps (3472), common service water pumps (7121), heat transport feed pumps (3331), moderator auxiliary pumps (3211)

Class C: Large machines with rigid foundation whose natural frequency exceeds machine operating frequency, e.g. moderator main circulation pumps (3211), boiler feed pumps (4323), condensate extraction pumps (4321), shutdown cooling pumps (3471), low pressure service water pumps (7131), circulating water pumps (7121)

Class D: Turbine-generator sets and other very large machines with operating frequency above foundation natural frequency, e.g. turbine-generator (4100), heat transport main pumps (3312)

\+ This height is measured from either the foundation level or any rigid structural support (e.g. thrust block), whatever is higher from the floor, to the measurement point.

\# Not applicable to Class D

Footnote: It is recommended that in addition to zero-to-peak levels, rms values be reported. The rms quantity is used to represent vibration severity in some standards. It also has value in diagnosis of vibration problems.

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